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# THE INTERNAL COMBUSTION ENGINE

# The Internal Combustion Engine

## Being a Text Book on Gas, Oil and Petrol Engines, for the Use of Students and Engineers

By

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# PREFACE

The Internal Combustion Engine is of such obviously growing importance that its study has become a necessity. Everywhere one finds evidence of the gradual replacement of steam plant, particularly in those cases where power users are in a position to avail themselves of the superior economy in moving and standing charges of the suction producer and gas engine. In marine propulsion the position of the steam engine is at present almost unassailed, but even there the situation is beginning to change.

It is remarkable that there should be no English text-book on the subject of the Internal Combustion Engine. A few short chapters dealing with it have often been included in text-books on the steam engine, but the subject has now become so important as to demand individual and exclusive attention. The present book is an endeavour to fill this gap. It deals with subjects in the borderland between the several allied sciences (notably physics and chemistry) and the exclusively practical sides of their application. It is hoped therefore that the student will be helped to understand something of the applications of those heat engines which work on the internal combustion principle, and the practical engineer to a better realization of the scientific principles concerned in the design and working of gas, oil and petrol engines. In order to economize space, and since it has been amply dealt with by many other writers, little is said of the historical side of the subject, nor has it been possible to include any discussion of the theory of the gas turbine. The treatment is necessarily mathematical in certain parts, but it requires nothing more than average acquaintance with mathematics, particularly if the reader who goes through the book for the first time omits the portions printed in small type. Acquaintance with the elements of the calculus is now so widespread that space has not been taken up by the adoption of methods which would avoid

its use. The introduction, into the theoretical treatment of the subject, of the principle of the now recognised variability of specific heats with temperature has involved the breaking of much new ground, so that it is impossible to expect complete success in avoiding mistakes and slips in the mathematical calculations. I shall therefore be very glad to have brought to my notice any corrections that may be found necessary.

Scattered throughout this volume will be found a number of problems for solution. They are chiefly drawn from the examination papers of the Board of Education and the Royal College of Science (Imperial College of Science and Technology) with both of which the author has had experience as an External Examiner. A number have also been taken from the papers set for the Mechanical Sciences Tripos at Cambridge.

In writing this book so many original papers and treatises have had to be consulted that it is not easy to make the requisite and proper acknowledgments. First, however, it is a pleasure to me to acknowledge my very great indebtedness to Professor Perry, to whom, as a student many years ago, and on numberless occasions since, my thanks are due for guidance, counsel and help generously placed at my disposal. I have also to thank Mr. Dugald Clerk and Professor Hopkinson for very kindly sending me copies of their valuable papers. I am indebted also to Mr. H. L. Burrell, Assoc. M. Inst. C.E., for checking the mathematical calculations and for working out the examples. For the illustrative matter I have to thank the Institutions, Firms and individuals mentioned in the following list, but chiefly my friend Mr. F. Strickland and Messrs. Chas. Griffin & Co. for permission to reproduce certain illustrations from their excellent treatise on "Petrol Motors and Motor Cars." Finally I have to tender my thanks to the Editors of *The Engineer* and *Engineering* for permission to reproduce certain parts of articles contributed to their columns.

H. E. W.

CHELSEA,
13*th August*, 1908.

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## ERRATA

PAGE

vii. Add "The Editor of *The Commercial Motor*".

xi. For line 2 read:—

$$\begin{cases} 1 \text{ yard} = 91{\cdot}44 \text{ cm.} \\ 1 \text{ metre} = 39{\cdot}37 \text{ inches.} \end{cases}$$

xiii. For $\begin{cases} C_p = \text{specific heat at constant pressure.} \\ C_v = \text{specific heat at constant pressure.} \end{cases}$

Read $\begin{cases} C_p = \text{specific heat at contant pressure.} \\ C_v = \text{specific heat at constant volume.} \end{cases}$

20. For $\Sigma + .\delta\phi$ read $\Sigma T.\delta\phi$

32. Lines 2 and 4. For "h.t.u." and "h.h.p." read "b.t.u." and "b.h.p."

64. At head of table, for $\frac{p}{V}$ read $\frac{\delta p}{\delta V}$

69. In line 6, for $\frac{1}{J}p.dV$ read $\frac{1}{J}p.\delta V$

77. In line 10, for "correction" read "convection". In line 28, for "always" read "already".

102. In Equation (3), the $\theta$, C and sin should all be on the alignment of the dots that follow the equation.

216. In line 7, for "hydroscopic" read "hygroscopic".

246. To title of Fig. 80, add "By the courtesy of the Editor of *The Commercial Motor*."

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# TABLE OF CONSTANTS.

1 inch $=25\cdot4$ millimetres or mm.
1 yard $=39\cdot37079$ inches.
1 litre $=1$ cubic decimetre $=1{,}000$ cc.'s, and 1,000 cc.'s of water weigh one kilogram.
1 cubic metre $=1{,}000$ litres.
1 cubic centimetre of water weighs one gram.
1 gallon $=0\cdot1605$ cu. ft. $=10$ lb. water at 62° F.
1 knot $=6{,}080$ ft./hour.
Weight of 1 lb. in London $=445{,}000$ dynes.
1 lb. avoirdupois $=7{,}000$ grains $=453\cdot6$ grams.
1 cu. ft. water weighs $62\cdot3$ lb.
1 cu. ft. air at N.T.P.—Normal temperature and pressure (0°C. and 760 mm.)—weighs $0\cdot0807$ lb.
1 cu. ft. hydrogen at N.T.P. weighs $0\cdot00557$ lb.
1 ft.-lb. $=1\cdot3562\times10^7$ ergs.
1 h.p.-hour $=33{,}000\times60$ ft.-lb.
1 electrical unit $=1{,}000$ watt-hours.
Watts $=$ volts $\times$ amperes.
1 h.p. $=33{,}000$ ft.-lb./min. $=746$ watts.
1 atmosphere $=14\cdot7$ lb./in.$^2=2{,}116$ lb./ft.$^2=760$ mm. of mercury $=10^6$ dynes/cm.$^2$ nearly.
A column of water $2\cdot3$ ft. high corresponds to a pressure of 1 lb./in.$^2$.
Absolute temperature $T=273\cdot7^\circ+\theta^\circ$ C.
$\pi=3\cdot1416$ or roughly $\frac{22}{7}$.
1 radian $=57\cdot3$ deg.
To convert common into Naperian Logarithms multiply by $2\cdot3026$, in other words $2\cdot3026\times\text{Log.}_{10}x=\log_\epsilon x$.
$\epsilon=2\cdot7183$.
$g=32\cdot182$ ft./sec.$^2$ or 981 cm./sec.$^2$ at London.
1 gallon $=4\cdot537$ litres.
1 cu. ft. $=0\cdot0283$ cu. metre.
1 lb./in.$^2=0\cdot0703$ kg./cm.$^2$.
1 inch water $=0\cdot187$ cm. mercury.
1 kg./cm.$^2=14\cdot22$ lb./in.$^2$ or nearly one atmosphere.
1 metric H.P. $=0\cdot986$ English H.P. $=75$ kg. metres/sec.

1 C.H.U. = 1 lb. water raised 1° C.

1 calorie = 1 kg. water raised 1° C. = 2·204 C.H.U. or 3,088 ft.-lb.

$\frac{°F.-32}{9} = \frac{°C.}{5}$; 1 B.T.U. = 1 lb. water raised 1° F.

1 H.P. = 10·68 calories/min.

1 cu. inch = 16·387 c.c.'s.

1 cu. ft. = 28·31 litres.

1 cu. inch cast iron = 0·26 lb.

,, wrought iron = 0·28 lb.

,, steel = 0·28 to 0·29 lb.

,, copper = 0·32 lb.

1 erg. = 1 dyne × 1 cm.

1 gramme-centimetre = 981 ergs.

1 ft.-lb. = $1{\cdot}356 \times 10^7$ ergs.

Energy obtainable from—

1 lb. coal = 12,000,000 ft.-lb.
1 lb. paraffin = 18,000,000 ft.-lb.
1 lb. petrol = 15,000,000 ft.-lb.
1 cu. ft. coal gas = 550,000 ft.-lb.

E, or Young's modulus for iron or steel = $30 \times 10^6$ lb./in.$^2$.

1 cu. metre water = 1 metric ton = 2,204 lb.

1 H.P.-hour = 641·4 calories.

## MOLECULAR WEIGHTS AND DENSITIES OF GASES.

| Gas. | Formula. | Molecular Weight. | Density (Air = 1). |
|---|---|---|---|
| Air . . . . . . . | — | — | 1·0000 |
| Carbon dioxide . . . | $CO_2$ | 43·90 | 1·5197 |
| Carbon monoxide . . | $CO$ | 27·94 | 0·9671 |
| Ethylene . . . . . | $C_2H_4$ | 27·95 | 0·9674 |
| Methane . . . . . | $CH_4$ | 15·97 | 0·5530 |
| Oxygen . . . . . . | $O_2$ | 31·93 | 1·1052 |
| Water vapour . . . | $H_2O$ | 17·96 | 0·6218 |
| Hydrogen . . . . | $H_2$ | 2·00 | 0·0692 |
| Nitrogen . . . . | $N_2$ | 28·02 | 0·9701 |

# TABLE OF CHIEF SYMBOLS USED.

$p$=pressure in lb./in.$^2$.
$V$=volume in cu. ft.
$T$=temperature absolute (Centigrade).
$\theta$=temperature as read by thermometer (Centigrade) in thermal calculations, and angle moved through in geometrical calculations.
$H$=heat energy in C.H.U. (Centigrade heat units).
$\phi$=entropy in ranks.
$v$=velocity in ft./sec. or miles/hour as the case may be.
$C_p$=specific heat at constant pressure.
$C_v$=specific heat at constant pressure.
$J$=mechanical equivalent of heat or "Joule's equivalent." (It may be written as 778 ft.-lb. in Fahrenheit scale or 1,400 ft.-lb. in Centigrade scale.)
$\alpha$=the constant term in the equation $C_p=\alpha+s\theta$.
$\beta$=the constant term in the equation $C_v=\beta+s\theta$.
$\alpha_1$=the constant term in the equation $C_p=\alpha_1+sT$.
$\beta_1$=the constant term in the equation $C_v=\beta_1+sT$.
$s$=the rate of change of specific heat with temperature as shown in above equations.
$t$=time in seconds.
$\eta$=efficiency.
$\gamma=\frac{C_p}{C_v}$.
$\gamma_1=\frac{\alpha_1}{\beta_1}$ and $\gamma_0=\frac{\alpha}{\beta}$
$R=C_p-C_v$.
$w$=weight of unit volume.
$k$=conductivity for heat
$K$=calorific valve.
$\omega$=angular velocity.
$I$=moment of inertia.
$s_1$=specific heat of metal.
$n$=the power in $pv^n$ equation.
$C$=amplitude of temperature variation in metal skin.
$M$=mass.
$E$=intrinsic energy.
h.p.=horse-power.
b.h.p.=brake horse-power.
K.W.=Kilowatts.

## CHAPTER I

# Introductory

**1.** Lecturing recently at the Royal Institution on "Flame in Gas and Petrol Motors," Mr. Dugald Clerk pointed out that although there are now in use stationary gas engines to the extent of over 2,000,000 h.p. and motor car engines to the extent of yet another 1,000,000 h.p., **very little is known as to the actual properties of the working medium employed.** This condition resembles that prevailing in the world of electricity, in which, notwithstanding the manifold uses to which the electric current is put, no one knows what an electric current is, or even what electricity itself is. At the first sight, therefore, it seems a hopeless task to write any "theory" of either the one science or the other; apparently there is no foundation at all to build on, and it is as futile a task as the building up of the walls of a house without any proper solid foundation being available. Let us pursue this analogy a little further. Dwellers in cities are now familiar with the modern method of house construction, which starts at the top and builds downwards, or starts in the middle and builds both upwards and downwards. The few necessary steel or ferroconcrete ribs and bones are there, and the flesh of walling is fitted on to them. This filling in can be started at any point, and the house assume a habitable aspect after it has proceeded long enough. Thus it is with the study of internal combustion engines and with electrical machinery. In the case of the latter we start with a definite relationship between the volt, the ampere and the ohm, and from this a whole mass of useful deductions and calculations can be made—even although no one knows really what a volt or an ampere or an ohm actually is. But then it is not

necessary to know the basis of the constitution of matter—as exemplified, let us say, in a cricket ball—in order to be able to play cricket. Indeed it is a recognized fact that those who know most about the constitution of matter are not those who play cricket best.

With the internal combustion engine we are placed in a parallel position: We do not know the way in which the **properties of gases vary with temperature,** and this coming on top of the fact that we do not know what a gas is, nor what temperature is, would seem to make progress somewhat heavy if not impossible. But as with the game of cricket, so it is here. We can find out some relationships between different properties of the substances we are using, and then if these relationships are sufficiently simple (as luckily they usually are), we can deduce from them a working theory. The more we calculate and the farther we get along the path the more necessary it is from time to time to check our position by direct experiment. If we get confirmatory results we go on, but if the facts fall outside our theory we must go back until we return to the last point at which experiment had coincided with theory, and then try again. This very necessary check is not always applied—and even when it is, experimenters not infrequently give the experiment but little chance of showing that they are wrong. In this way theories have often been put forward which have had afterwards to be abandoned.

A theory on which practically all engineers, if over thirty years of age, were brought up, was that the specific heats of gases were *constant*, or sufficiently nearly so for all practical purposes. Now, thanks to the work of MM. Mallard, Le Chatelier, Holborn, Austin, Hopkinson, Dugald Clerk, Burstall and others it has been shown that this is not so. It has therefore become necessary to work out a new theory of gas engines based on this latest information. This is one of the chief reasons why this book has been written. It is unfortunate that the fact of the specific heats of the gases employed increasing with temperature should make the calculations more difficult instead of less difficult, but the problem exists and it must be faced. Calculations and

theories should be built on true foundations, or not built at all. No full theory has yet been put forward. The author, however, makes an attempt at a beginning, and no better piece of work could be given to a student than to take the theory as the author leaves it and carry it on farther and farther. He must not, however, forget to check his steps by constant experiment, and test them as mentioned above.

2. With these preliminary observations we will consider what are the **sources from which is obtained the energy** that drives our engines. They are:

1. Solar heat (past or present).
2. Tidal Action.
3. Molecular action (radio-activity).

**Solar** heat is available in two ways—either in the form of stored energy, or in the form of energy pouring in hour by hour and day by day. The former is best exemplified by coal and oil. These substances have been formed by solar heat acting on the earth through many thousands, tens of thousands, hundreds of thousands and even millions of years. When we burn oil or coal we are therefore spending capital. The solar radiation that is received each day by the earth is little used directly, although gigantic mirrors and boilers have been erected in some parts of the world, such as Mexico; but although the heat is thus obtained for nothing, the apparatus required is very costly. Indirectly the Sun's rays are used in windmills and waterfalls, as it is simply the heat of the Sun acting differentially over the earth's surface that causes the wind, and the evaporative power of the Sun's rays that leads to clouds being formed and therefore to rain, streams and waterfalls.

**Tidal** action, which is jointly due to the Moon and the Sun, and chiefly to the former, is little used at present owing to the cumbrous mechanism involved, and the chance of damage through storms. But it remains a possibility.

Lastly we come to **Molecular** action, or **Radio-activity**—the last discovered, and the one most full of promise for the future. As illustrative of its marvellous potentialities the following extract may be given from a paper by Sir Oliver

Lodge in the *Philosophical Magazine* * on "The Density of the Ether." Speaking of the energy locked up in the ether:—"This is equivalent to saying that 300,000,000,000,000,000 kilowatt hours, or the total output of a 1,000,000 k.w. power station for 30,000,000 years, exists permanently, and at present inaccessibly, in every cubic millimetre of space." Also in his 1908 Royal Institution lecture, after saying that the mass, momentum and kinetic energy of matter were really those of the ether, he restated his estimate thus—"the intrinsic energy of 1 cu. mm. of space was equivalent to 1,000 tons moving with the velocity of light. That would mean an output of a station of 1,000,000 h.p. working for 40,000,000 years." Any figures of this kind must of course be looked upon as very rough guesses, but it is undoubtedly the case that **enormous stores of energy** are contained even in the smallest particles of matter. It is difficult to get at it, and fortunate it is that it is so difficult, else catastrophic explosions might ensue from the simplest experiments. The phenomenon presented by radium shows however that this energy does sometimes leak very gradually away, and by the time that the leak is understood our present notions about the physical world may have to undergo a revolutionary change. The student may now ask: Why this trouble to learn about theories which may all prove to be fallacious? The reply is that it is by studying them that he will himself be the better prepared to solve the problems which will arise in the future, and that in any case the better known theories of the present day have been compared with experimental results and conform to them very closely. It may be, of course, that they are sometimes little else than empirical laws or deductions from a brief series of experiments, but even so they are of use, and much has been done in certain fields of engineering work by such theoretical investigations: As witness the work of Clerk Maxwell and Hertz in electrical work, and Rayleigh in the theory of sound. Theoretical and practical work should always go hand in hand, and

* April, 1907.

whenever it happens that they do not do so, difficulties arise from the one side or the other.

The expansion of scientific knowledge is to be desired most earnestly from the point of view of an advancing civilization. Sir W. Besant in his *Westminster* remarks :—" The vanished civilization of Roman Britain was very far superior to anything that followed for a good deal more than 1,000 years." It vanished because there was not in that high degree of civilization sufficient knowledge of the science of machines to repel the barbaric invader. Our present civilization could only be overthrown physically by a still higher scientific civilization, and though we think that even that would be a great loss, it does not follow that the future ages would be of the same opinion. It is no exaggeration to say that prominent among the civilizing factors of our time is the **internal combustion engine.** To realize this we must consider the many uses to which it is put.

**3.** The difference between a gas engine and a steam engine is that the former is an *internal* combustion engine, and the latter is not. In a steam engine combustion takes place in the furnace of a boiler, i.e. *external* to the working cylinder altogether. In a gas engine, or oil or petrol engine, however, the combustion or burning of the fuel takes place *in* the working cylinder. Hence the name **Internal Combustion Engine.** The most popular recent development of the internal combustion engine is the petrol motor car—many tens of thousands have been built in the last few years. Another variety at the other end of the scale is the gun. All cannons, guns, rifles and pistols are really internal combustion engines, and in part at least similar considerations govern their design. Another form of internal combustion engine which is rapidly becoming familiar is the large gas engine, operating on waste gases from iron furnaces, or coke ovens, or on gases specially prepared in gas producers.

It is not easy to find any reliable figures as to the progress made in this country, but the United States Government has produced through the medium of its Census Bureau some very striking statistics as regards the use of power

for industrial purposes in the United States. In 1870 the total power employed in the country was 2,346,000 h.p. In ten years it increased to 3,411,000 h.p.; ten years later to 5,955,000 h.p. By the end of the century (i.e. in ten further years), it was 10,410,000 h.p. And in 1906 it had grown to 14,465,000 h.p. Before 1890, the main sources of power were steam and water, but since then a great advance has been made in the utilization of gas power. Between 1890 and 1900 the use of gas power increased by *1,400 per cent.* And between 1900 and 1905 the total output more than doubled.

Students will now be in a position to grasp the magnitude and importance of the problems which will be dealt with in the succeeding chapters.

# *SECTION I*

# THEORY

## CHAPTER II

# Thermodynamic Cycles

Manner of Working of Internal Combustion Engines—Units—Perfect Gases—Isothermal Expansion—Adiabatic Expansion—Entropy—Constant Temperature Cycle—Constant Pressure Cycle—Constant Volume Cycle—Thermal Efficiency.

**4. Manner of working of Internal Combustion Engines.** The popularity of the internal combustion engine is now so marked that almost every one who is at all interested in engineering work is familiar with its method of working. The internal combustion engine, whether in the guise of gas engine, oil engine, or petrol engine, is to be found everywhere; as an instance the writer recently noticed a small suction plant and gas engine located at the top of a mountain in Germany and working quite smoothly, although no attendant was to be found near it.

The working of a steam engine is well known. It is usually double acting and every outward stroke of the piston is a working stroke and every return stroke is what would in gas engine parlance be called a scavenging stroke. In an internal combustion engine this is all changed, and in the most familiar type air is taken into the gas engine cylinder (and with it some gas, oil or petrol * in order to heat the air by explosion and so allow work to be done by expansion) on the outward stroke of the piston, compressed on its return stroke, and explosion is then made to occur by an electric spark. The exploded mixture then

* In America paraffin is called kerosene, and petrol is known as gasoline.

expands and does work on the outward stroke, whilst on the final return stroke it is ejected from the cylinder. This process is then repeated. **Four strokes therefore go to a working cycle** (which is therefore often called a four-stroke cycle) instead of two as in a single acting steam engine,* so that for the same maximum pressure a gas engine would require to be twice as big as a steam engine, both being single acting, to give the same power; but actually the maximum pressure in a gas engine is far higher than in a steam engine, so that the power is not so disproportionate to the size. Most of the older gas engines were single acting; that is to say, the pressure was allowed to act on one side of the piston only, but there are now many double acting gas engines made, and, as this enables the diameter of the cylinder to be reduced, it means an increase in the power of the engine for a given weight. The four-stroke cycle above described is the famous **Otto Cycle.** Many years ago Mr. Dugald Clerk invented a two-stroke cycle in which the suction and compression were done in another chamber and not in the working cylinder. This is called the **Clerk Cycle,** and it enables every outward stroke of the piston to be a working stroke just as in the steam engine; and if the gas engine also be made a double acting one the proportion of working strokes becomes exactly the same as in a steam engine, and problems of uniformity of torque and balancing can be solved in just the same way.

**5.** There are therefore **two possible cycles** on which the internal combustion engine as at present devised can work, either on the Otto or the Clerk cycle. When one of these cycles has been selected to work with, there are three variations in the procedure which may be adopted. The heat liberated by the chemical combination which occurs on explosion may be so utilized that it is given to the gaseous mixture (which is always chiefly air) either at constant temperature, at constant pressure, or at con-

* A single acting engine is one in which pressure is admitted to one side of the cylinder only. In a double acting engine it is admitted to both sides.

stant volume. Any one of these **three** methods (they also are called cycles, thus the "constant-pressure cycle") can be used, giving six possible combinations, of which the Otto constant-volume cycle is the most common, whilst the Otto or Clerk cycle in combination with the constant-temperature cycle would be most efficient for a given temperature range. The nearest approach to the latter at present is probably found in the Diesel oil engine, which, however, still more closely follows the constant-pressure cycle.

It is now necessary to examine each of these three ways of adding heat to the gaseous mixture, in order to see how the efficiency is effected.

The remarkable fact will be proved that for each and all of these cycles the **maximum possible thermal efficiency** is equal to an expression which depends alone upon the **ratio of compression** employed on the compression stroke. By ratio of compression is meant the ratio of the **volume** of the cylinder when the piston is as far out as it goes, to the volume of the cylinder when the piston is right in, i.e. $\dfrac{\text{cylinder volume}}{\text{clearance volume}}$. Before proceeding to prove this, it will be necessary to define our units.

**6. The unit of heat** is the heat required to raise 1 lb. of water through 1° Centigrade or Fahrenheit. The former is called a "Centigrade heat unit" or C.h.u., and the latter a "Fahrenheit heat unit," or sometimes a "British Thermal Unit" or B.T.U. The equivalent unit in the metric system is the Calorie, or the heat required to raise 1 kilogram of water through 1° C. One Calorie = 2·204 C.h.u.

**The unit of energy or work** is the foot-pound, being the work done in raising 1 lb. weight one foot high.

Dr. Joule of Manchester was the first to measure the number of foot-pounds of energy that were equivalent to one heat unit. His measurements have subsequently been revised and the following figures probably represent the correct results as nearly as possible. The number of foot-pounds of energy equivalent to one heat unit is called **Joule's Equivalent.** If the heat unit be a Centigrade one, the

value of the equivalent is 1,400 ft.-lb., but if a Fahrenheit one, its value is 778 ft.-lb. Calculations are often made in both Centigrade and Fahrenheit heat units, so that both values must be remembered. Their ratio is of course $\frac{9}{5}$.

Continental engineers, and some in this country and America, work with the metric units, and the unit of energy then becomes the kilogram-metre. The relative value of the different constants is given in the table of constants at the beginning of the book.

**7.** We know by **Boyle's Law** that provided the temperature be kept constant the volume of a mass of gas will vary inversely as the pressure. This may be written $p.V. =$ constant where $p$ is the pressure and $V$ the volume. Also we know by **Charles's Law** that under constant pressure equal volumes of different gases increase equally for the same increase in the temperature. Further, that if a gas be heated under constant pressure, equal increments of its volume correspond very closely to equal intervals of temperature as measured by a mercury thermometer. It is found by experiment that the amount by which a gas expands when its temperature is changed by one degree Centigrade, pressure being kept constant, is about $\frac{1}{273}$ or $\frac{1}{274}$ of its volume at 0 °C. From this it follows that if the gas obeyed this law at all temperatures it would have contracted into half its volume at —137 °C. and to no volume at all at —274 °C. The latter point (—274 °C.) is called the **absolute zero of temperature,** and it is often used as a starting point from which to measure temperature. In that case the temperatures so measured are called **absolute temperatures** and equal the ordinary temperature plus 274 when working on the Centigrade scale or plus 461 for the Fahrenheit. Combining these two laws a **perfect gas** can be defined as one which follows the law

$$\frac{p.V.}{T} = \text{constant}$$

where $p$ is the pressure (absolute), $V$ is the volume and $T$ is the temperature (absolute) ; all of these being measured on any scale or system that may be found convenient, with the proviso that whatever system is adopted for a given calcu-

lation must of course be adhered to throughout. The ordinary gases, such as hydrogen, oxygen and nitrogen, do not follow this equation quite exactly, but they do so very nearly, and even a gas such as $CO_2$, which being relatively easily liquifiable at low temperatures does not follow it so closely, is sufficiently near to it at the temperatures and pressures met with in gas engines to enable it to be used without sensible error. Students of the steam engine will remember that if $p$, $V$, and $T$ are measured on the usual system of units, and the working mass of gas be taken as 1 lb., the value of the "constant" is $(C_p—C_v)J$ and is commonly indicated by the letter $R$, so that

$$pV=RT=J(C_p—C_v)T$$

where $J$ is, as usual, Joule's equivalent, and where $C_p$ is the specific heat at constant pressure (i.e. the number of heat units required to raise 1 lb. weight of the gas through 1° Cent. when kept at constant pressure), and $C_v$ is the specific heat at constant volume (i.e. the number of heat units required to raise 1 lb. weight of the gas through 1° Cent. when kept at constant volume).

**8.** It is not difficult to show that the constant $R$ **must** be equal to $(C_p—C_v)J$. Consider a volume of gas (at $p_0$, $V_0$ and $T_0$) confined in a cylinder of exactly one square foot in cross sectional area (i.e. about 13½ inches in diameter), and having a weightless piston above it to keep it in. Let the temperature increase to $T_1$ and the volume to $V_1$ keeping the pressure constant and equal to $p_0$. Then the heat units supplied to the gas must clearly be equal to $C_p\ (T_1—T_0)$. Part of this heat goes to heat up the gas and the other part to do the external work of expanding from volume $V_0$ to volume $V_1$ against a pressure of $p_0$. Now the heat units used to heat up the gas are equal to the heat that would be required for the whole operation had no expansion been permitted,* in which case the heating would have been done at constant volume and the heat units required equal to $C_v(T_1—T_0)$.

It follows therefore that the difference between $C_p(T_1—T_0)$ and $C_v(T_1—T_0)$ must be equal to the external work done,

* See p. 16.

measured in heat units, i.e. to $p_0(V_1-V_0) \div J.$, since the load on the piston is $p_0$ (that is a pressure of $p_0$ lb. per sq. foot), acting on an area of 1 sq. ft. and the vertical motion of the piston in feet is $(V_1-V_0)$.

Therefore $C_p(T_1-T_0)-C_v(T_1-T_0)=\frac{p_0}{J}(V_1-V_0)$

or $$(C_p-C_v)(T_1-T_0)=\frac{p_0}{J}(V_1-V_0)$$

or $$p_0\frac{V_1-V_0}{T_1-T_0}=J(C_p-C_v) \qquad (1)$$

But $$\frac{p_0V_0}{T_0}=\frac{p_0V_1}{T_1}$$

or $$\frac{V_0}{T_0}=\frac{V_1}{T_1} \qquad V_1=T_1\frac{V_0}{T_0}$$

Substitute in (1) and

$$p_0\frac{T_1\frac{V_0}{T_0}-V_0}{T_1-T_0}=J(C_p-C_v)$$

or $$\frac{p_0V_0}{T_0}=J(C_p-C_v)$$

But $$\frac{p_0V_0}{T_0}=R$$

therefore $R=J(C_p-C_v)$.

**9.** The equation

$$\frac{pV}{T}=R=J(C_p-C_v)$$

is true for all values of $p$, $V$ and $T$. If equal weights of two different gases be taken and in both the $p$ and $T$ are adjusted to be the same, the volumes will be in the inverse ratio of their respective densities. Thus 1 lb. of hydrogen will occupy a far larger space than 1 lb. of oxygen, but it follows from the above that an equality between the two gases must exist as regards the following expression.

$$(C_p-C_v) \times \text{density}$$

and the following table illustrates this in practice—

| Gas. | $C_p$. | $C_v$.* | Density relative to Air. | $(C_p—C_v)\times$ density. |
|---|---|---|---|---|
| $H_2$ . . | 3·4090 | 2·4060 | 0·0692 | 0·0694 |
| $N_2$ . . | 0·2438 | 0·173 | 0·970 | 0·0687 |
| $O_2$ . . | 0·2175 | 0·155 | 1·105 | 0·0691 |
| $CO_2$ . . | 0·217 | 0·171 | 1·520 | 0·0700 |

The fact that there are any differences at all is because these gases are not absolutely "perfect gases." The assumption is implied moreover that the specific heat is absolutely independent of temperature, and although for many calculations this is sufficiently nearly true, there are others, as will appear in a subsequent chapter, in which this is by no means the case.

In problems connected with the internal combustion engine, one is continually coming across the symbols $C_p$ and $C_v$ and it is necessary at the earliest stage to get thoroughly familiar with their use.

**10.** Another form in which they constantly recur is as **the ratio** $C_p \div C_v$, and this ratio is known by the Greek letter $\gamma$, **(gamma)**, so that

$$\frac{C_p}{C_v}=\gamma$$

and since

$$C_p—C_v=R$$

$$\frac{C_p}{C_v}—1=\frac{R}{C_v}$$

$$\gamma—1=\frac{R}{C_v}$$

The expression $(\gamma—1)$ is constantly recurring. The value of $\gamma$ is usually from 1·3 to 1·4, the latter being the value for air.

It has been said that

$$\frac{pV}{T}=R$$

* The student of physics will remember that Dulong and Petit showed that the product of $C_v$ by molecular weight is practically constant for all gases.

and that $R$ is a constant for the gas concerned. If the temperature be kept constant

$$pV = RT = \text{constant}$$

and this is sometimes called the hyperbolic law of expansion. In reality there is no such *law*, and the statement is only a variation or simplification of the perfect gas formula. If however the gas really does expand according to the equation

$$pV = \text{constant}$$

it will be doing external work and inasmuch as the energy it contains will not alter, seeing that the temperature does not change, heat must be being supplied to the gas at a rate equal to that at which the external work is being done. Suppose, however, that the conditions had been such that the gas did not receive or lose heat, and yet was *compelled* to expand or contract by reason of the piston being forcibly raised by some means or other; how would the pressure and temperature be affected ?

Now it was discovered by Dr. Joule that in all gas operations the heat $H$ supplied from outside must be balanced by **the gain in temperature of the gas plus the external work done,** so that

$$H = C_v.\delta T + p.\delta V$$

where $\delta T$ and $\delta V$ represent the changes of temperature and volume which occur in any transformation. It is most convenient to regard the process as split up into little steps and therefore to regard $\delta T$ and $\delta V$ as **small increments** of temperature and volume.

If we consider a case in which the gas is being neither allowed to gain nor to lose heat $H$ must be zero

$$\text{or } C_v.\delta T + p.\delta V = 0$$

and from this it is easy to show by means of the integral calculus that $pV^\gamma = \text{constant}$. This transformation is called an **Adiabatic** one because no heat is allowed to pass into or away from the gas.

For those who are acquainted with the calculus it may be added that the steps of the integration are

$$C_v \, . \, \delta T + p . \delta V . = 0$$

therefore $\dfrac{dT}{dV} = -\dfrac{p}{C_v}$ in the limit,

but $pV = RT = (C_p - C_v)T$.

or $\dfrac{dT}{dV}(C_p - C_v) = p + V . \dfrac{dp}{dV}$.

Substitute and

$$-\frac{p}{C_v}(C_p - C_v) = p + V . \frac{dp}{dV}.$$

or $-\gamma p = V . \dfrac{dp}{dV}$.

$\dfrac{dp}{p} = -\gamma . \dfrac{dV}{V}$ and integrate.

therefore $pV^\gamma = \text{constant}$,
and this is one of the most important equations that there are in gas engine work.

**11. Two fundamental gas formulæ** are—

**pV =constant** is an **Isothermal Expansion.**

**pV$^\gamma$=constant** is an **Adiabatic Expansion.**

The equation $pV^\gamma$=constant only gives the value of $p$ and $V$ for this transformation, but it is quite easy to get the value also of $T$, as the relation $\dfrac{pV}{T} = R$ holds good for **all** transformations of whatever sort or condition.

Thus, since $p = \dfrac{RT}{V}$,

we have $pV^\gamma = \dfrac{RT}{V} V^\gamma = \text{constant}$;

or $TV^{\gamma - 1}$=constant, and this gives the relation between $T$ and $V$. Again

$$V = \frac{RT}{p}.$$

So that $pV^\gamma = p\left(\dfrac{RT}{p}\right)^\gamma = \text{const.}$

or $\dfrac{T^\gamma}{p^{\gamma-1}} = \text{constant}$,

and this is the relation between $T$ and $p$.

If at the beginning of any transformation $p=p_0$, $V=V_0$ and $T=T_0$

Then

$$p_0V_0^{\gamma}=\text{a constant},$$
$$T_0V_0^{\gamma-1}=\text{a constant},$$
$$\frac{T_0^{\gamma}}{p_0^{\gamma-1}}=\text{a constant},$$

and therefore the pressure, volume and temperature at a succeeding state being called $p_1$, $V_1$, and $T_1$ it follows that

$$p_1V_1^{\gamma}=p_0V_0^{\gamma}$$

and

$$T_1V_1^{\gamma-1}=T_0V_0^{\gamma-1}$$

and

$$\frac{T_1^{\gamma}}{p_1^{\gamma-1}}=\frac{T_0^{\gamma}}{p_0^{\gamma-1}}$$

**12. As an example,** if a gas is compressed adiabatically in the ratio of 10 to 1, i.e. so that it only occupies $\frac{1}{10}$ part of its former volume, then the temperature will alter thus—

$$T_1V_1^{\gamma-1}=T_0V_0^{\gamma-1}$$

where

$$\frac{V_1}{V_0}=\frac{1}{10}$$

so that

$$\frac{T_1}{T_0}=\left(\frac{V_0}{V_1}\right)^{\gamma-1}=(10)^{\gamma-1}$$

and as

$$\gamma=1\cdot4 \text{ (say)}$$
$$\frac{T_1}{T_0}=(10)^{0\cdot4}=2\cdot51,$$

i.e. the temperature absolute increases by 151 per cent., so that had $T_0$ been 290 (i.e. a temperature of 16° C. to start with, about the temperature of a room in the summer time) then

$$\frac{T_1}{290}=2\cdot51$$
$$\therefore\ T_1=729^{\circ} \text{ abs.}$$

$\therefore$ resulting temperature $=729—274=455°$ C.

This explains the heating of the air which occurs when air is suddenly compressed as in pumping up a bicycle tyre. It is necessary to say "suddenly," as if done slowly the heat would have time to escape, and the change would be far from adiabatic, and would in the limit become isothermal.

**13.** The **entropy diagram** is familiar, if only in appearance, to all who have studied the Steam Engine and will therefore probably be already well known to many readers. The abstract definition of what entropy is makes a rather long and difficult study, but it must be remembered that close scientific definitions of even the commonest things tend to become abstruse and to suggest a strangeness which does not adhere to one's familiar conception of them. So it is with entropy—it is most easily described by a reference to its properties and uses. The two ideal types of the expansion of gases are the **isothermal** and the **adiabatic**. In the former the **temperature** remains a constant. In the **adiabatic** expansion the temperature varies, but what *does* remain constant is the **entropy** of the gas. That is to say the amount of entropy remains the same throughout adiabatic transformations. This is probably the simplest way there is of defining entropy. The measure of the entropy depends upon the point from which it is reckoned, just as a temperature reading varies according to whether it is measured from the absolute zero or from 0° C. In the case of entropy it is found most convenient to measure from the state at 0° C. The amount of entropy in a substance is calculated thus:

Given a mass of gas and given that a certain quantity of heat is put into it, how does the entropy change? This is, in its simplest form, the problem which has now to be faced.

If the certain quantity of heat is called $\delta H$ and the absolute temperature $T$, then the gain in entropy, which is usually called $\delta\phi$ is $\frac{\delta H}{T}$, so that

$$\delta\phi = \frac{\delta H}{T}$$

This is the mathematical definition of what is here called entropy.

It may be, however, that the temperature $T$ will vary during this change and if so the whole operation must be considered as divided into little parts each with its own temperature and the whole afterwards added up. The usual mathematical way of stating this is

$$\Sigma\delta\phi = \Sigma\frac{\delta H}{T}$$

or in the limit and using the notation of the calculus

$$\int d\phi = \int \frac{dH}{T}$$

To evaluate the entropy for any given change in the state of the gas it is necessary to know how $H$ depends on $T$. If in a simple case $H$ varied directly with $T$, say $H = aT$ (water for example) then the equation would be—

$$\int d\phi = \int \frac{dH}{T} = \int a.\frac{dT}{T} = a \int \frac{dT}{T}$$

or $$\phi = a \log_\epsilon T + \text{ constant},$$

and since by definition $\phi = 0$ when $T = 274$

it follows that the expression for the entropy is—

$$\phi = a \log_\epsilon \frac{T}{274}$$

and in this way its value could be found for any given temperature.

If a given volume of gas has its pressure, volume and temperature changed in any way so that after undergoing several such operations it returns to the same state, then the values of its entropy and temperature will, when plotted on a sheet of paper, form a closed curve which has the useful property that its area measures the heat supplied just as the area of the $P.V.$ or indicator diagram measures the mechanical work done. If measures are made in both cases in ft.-lb. both diagrams in any particular case would have the same area. That the $T.\phi$ diagram really does measure heat supplied by its area is evident, since by definition

$$\delta\phi = \frac{\delta H}{T}$$

or $$T.\delta\phi = \delta H$$

so that $$H = \Sigma T.\delta\phi$$

and $\Sigma +.\delta\phi$ is of course the area of the diagram. In Fig. 1 is shown a very simple entropy diagram. The gas starts at the point 1 ; the temperature is then increased to the point 2, whilst the entropy remains constant—an adiabatic compression ; then the gas has its temperature kept constant from 2 to 3, whilst the gas receives heat and the entropy increases from 2 to 3 ; then from 3 to 4 the gas expands adiabatically as the entropy is constant and the temperature falls to 4 ; then from 4 to 1 the temperature remains steady, whilst the gas gives up its heat and the entropy diminishes from 4 to 1, so bringing the gas back to its original state, and ready to go through the cycle again. This is the well-known **Carnot Cycle,** which is so often shown

on the $P.V.$ diagram but is so much more easily understood on the $T.\phi$ diagram. What is the area of the diagram in Fig. 1? It is plain from what has been said that the

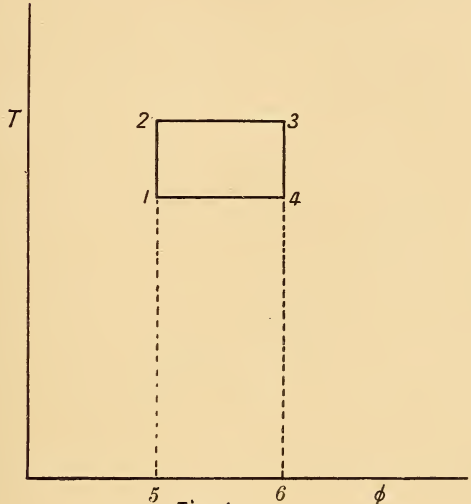



Fig. 1.

area 2, 3, 6, 5 represents **the heat taken in** by the engine, and the smaller area 1, 4, 6, 5 that rejected.

So that the efficiency or $\frac{\text{heat utilized}}{\text{heat supplied}} = \frac{\text{area } 1, 2, 3, 4}{\text{area } 2, 3, 6, 5}$

and this ratio is obviously equal to $\frac{2,1}{2,5}$

$$\text{i.e. to } \frac{\text{max. temp. of cycle—min. temp. of do.}}{\text{max. temperature of cycle}}$$

which is the customary expression for the efficiency of the Carnot Cycle. This is an instance of how simple the use of the $T.\phi$ diagram makes such calculations.

**14.** In this last named figure all the lines were parallel to one or other of the axes. This was because an ideal cycle of the simplest nature was being followed. In Fig. 2, the sloping lines $AB$ and $BC$ have been drawn at random. **What changes of state** would they represent?

The line $AB$ shows an increase of both entropy and temperature, both of them increasing at about an equal rate.

So that heat is being given to the gas, and the temperature is increasing meanwhile. This is generally similar to what goes on during explosion in a gas engine cylinder as the gas takes in heat from the effect of chemical combination, and the temperature rises while it does so. Having arrived at the point *B* the gas now follows the line *BC* during which the gas continues to take in heat, and the temperature decreases. This is what would occur, on a lesser scale, in a

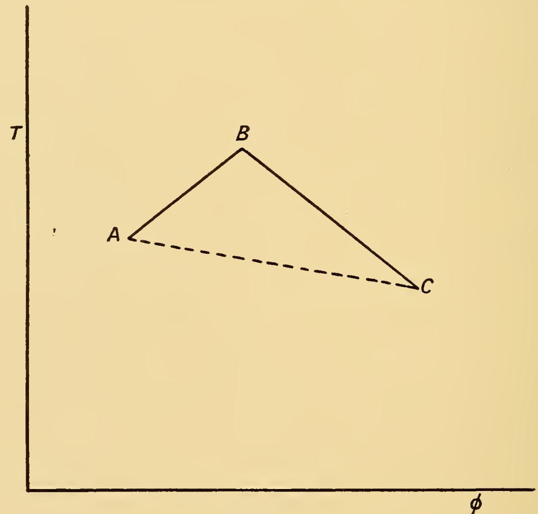



Fig. 2

gas engine cylinder were the combustion of the gas to continue right through the working stroke instead of ending at the point of highest temperature, as it is now generally believed to do. Then to get the gas back to its original state the line *CA* is followed, and during it the gas gives out its heat at a nearly steady temperature, i.e. almost an isothermal compression. No gas engine works exactly on this cycle, which was one drawn at random to show how any cycle whatsoever can be very easily and readily studied by the use of the $T.\phi$ diagram. It is obvious from the diagram that the efficiency of this triangular cycle would be a low one as the area is small having regard to the temperature variation represented.

Gas engine indicator diagrams are often turned into $T.\phi$ diagrams, but it is necessary that certain precautions

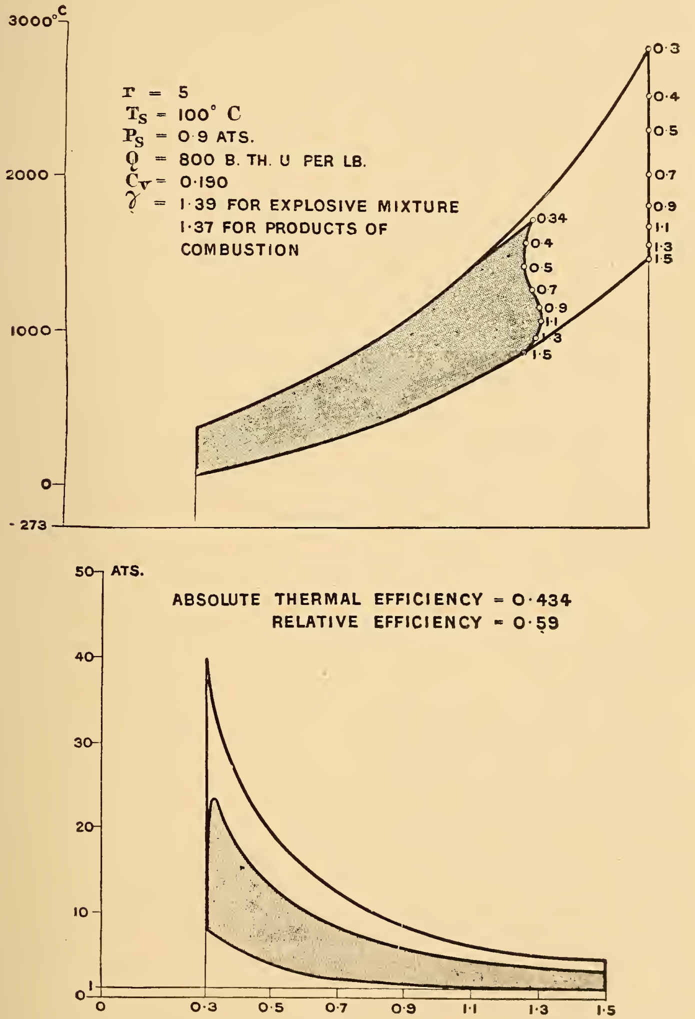



Fig. 3.

should be taken in doing so. The difficulty lies in the fact that the working fluid does not remain in the cylinder for a number of cycles, but is periodically discharged to exhaust,

and a fresh charge brought in. The cycle can, however, be treated as a continuous one if the exhaust gases are considered to have their relatively high temperature and pressure reduced to those of the incoming charge, the volume being kept constant. In an appendix to an Institution of Civil Engineers report*, Captain Sankey has shown a number of *P.V* and *T.φ* diagrams for the same gas engine cycles, and by the kind permission of the Council of the Institution, one of them is reproduced in Fig. 3. The fine lines show the *T.φ* and *P.V* diagrams for an ideal engine, whilst in the shaded portion is given the same diagrams for a probable *actual* engine. The wavy part of the shaded curve shows the expansion period of the cycle. It has been drawn to show the cooling of the gas to the walls and piston during the beginning of expansion, and the subsequent flow of heat in the reverse direction during the latter part of the stroke, this effect dying away again at the very end of the stroke, possibly on account of the slow motion of the piston at that point, which would allow the walls a greater amount of time in which to part with their heat.

Before dealing with the efficiencies of the various cycles of working it is necessary to say something about the working medium. The gaseous mixture that enters a gas engine (for oil or petrol engines the same considerations apply) is usually $\frac{9}{10}$ air and the rest gas, and even when the proportion of air is not quite so high as this, by far the greater part of the mixture is simply air. Air is in fact the working substance, and gases, oils and petrols are used merely to heat it up to the point required to carry out the predetermined cycle of operations. So that although the writer gives the thermal constants, not only for air but also for the other gases, etc., concerned, it must be remembered that air is the most important factor, and that inasmuch as air is $\frac{4}{5}$ **nitrogen,** it is the latter gas which is most concerned, however passively, in the working of internal combustion engines. The following table shows the other gases chiefly concerned in such operations when using various working substances.

* *I. C. E. Proc.* Vol. 162.

| | Town Gas. | Producer Gas. | Blast Furnace Gas. | Coke-Oven Gas. |
|---|---|---|---|---|
| | per cent. | per cent. | per cent. | per cent. |
| CO . . . . . | 7 | 20 | 25 | 8 |
| $CO_2$ . . . . . | 2 | 9 | 6 | 2 |
| H . . . . . . | 46 | 21 | 2 | 53 |
| N . . . . . . | 3 | 48 | 66 | 5 |
| Hydrocarbons . | 42 | 2 | 1 | 32 |
| B.T.U. per cub. ft. about . . | 600 | 150 | 90 | 540 |

**15. Ideal Standard Cycles.**—Every one who is acquainted with steam engines knows that the standards of comparison are the Carnot Cycle or the Rankine Cycle, that is to say, these two ideal cycles of operation are the standards by which actual engines are best judged. It would be unfair to complain of a steam engine that gave a thermal efficiency of 0·25 when that ideally possible for the temperatures employed was only 0·30, indeed such an engine must be greatly superior to any yet constructed, and although 25 per cent. efficiency does, it is true, mean that 75 per cent. of the energy is wasted, yet in reality the engine is a very good one as it yields $\frac{\cdot 25}{\cdot 30}$, i.e. 83 per cent. of what is ideally possible. It is this figure of 83 per cent. which should really be looked to. The figure of 0·25 gives little information indeed, but the figure of 83 per cent. shows at once that unless the manner of working be altogether changed there is only 17 per cent. left to improve upon. In a steam engine the endeavour is to keep the cylinder hot and so prevent the condensation which causes the efficiency to fall below its possible level. In a gas engine, on the contrary, the endeavour is to cool by the cylinder to keep the engine from jambing and otherwise working badly. Clearly there is here a marked difference in operation, and correspondingly it becomes necessary to devise new standards of comparison suitable to the working of gas engines.

There are **Three Ideal Standard Cycles**, viz.—

1. The constant temperature type.
2. The constant pressure type.
3. The constant volume type.

Each of these has been investigated by a Committee appointed by the Institution of Civil Engineers, and as it is desirable to avoid a multiplicity of methods of dealing with the same thing, the author will follow generally the procedure they recommend.

**16. The Constant Temperature Type.**—In an engine of this type, all the heat is taken in at the highest *temperature* and *all* is afterwards rejected at the lowest *temperature.* Those who have followed what has been written in this chapter will recognize this as the Carnot Cycle, and it can be proved that for the same temperature limits no possible treatment of a heat engine can give a higher efficiency than

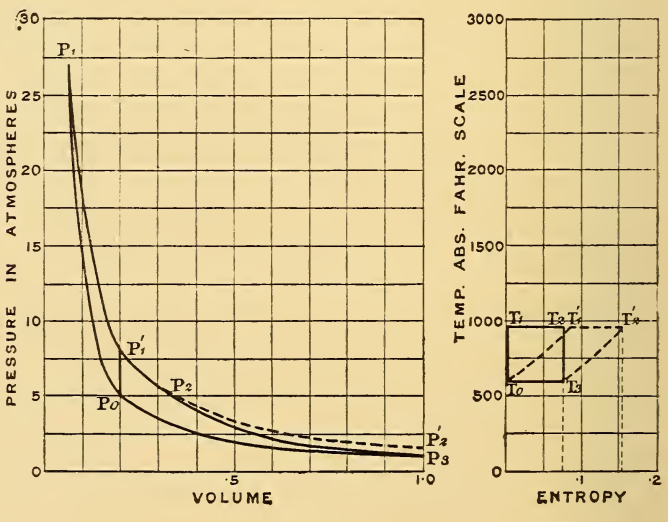
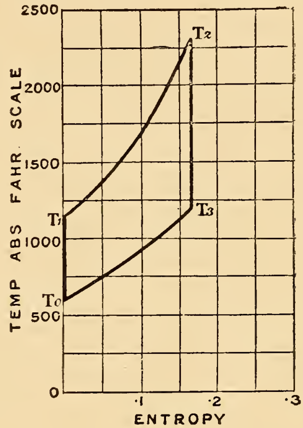



FIG. 4.

is theoretically obtainable in this way. The diagrams in Fig. 4 show at once that the efficiency is $\frac{T_1 - T_0}{T_1}$ where $T_1$ is the highest temperature and $T_0$ the lowest, both of course being reckoned from the absolute zero of temperature. $T$ is **always** used in this book to mean temperature absolute, and $\theta$ to mean temperature as read on a thermometer. The diagram above referred to is of course an entropy diagram,

commonly referred to as a "$\theta,\phi$" diagram, but as $\theta$ is being kept for temperatures which are not absolute it would be better to say "$T,\phi$." A "$P.V$" diagram is also shown and any one at all acquainted with the working of steam or gas engines will at once recognize that for any given h.p. the cylinder would require to be exceedingly large and costly, so that the extra economy in the matter of coal, brought about by its high efficiency would be more than counterbalanced by the inconvenience of the size of the engine and by the extra annual outlay necessary to provide for interest and depreciation on the enhanced capital cost.



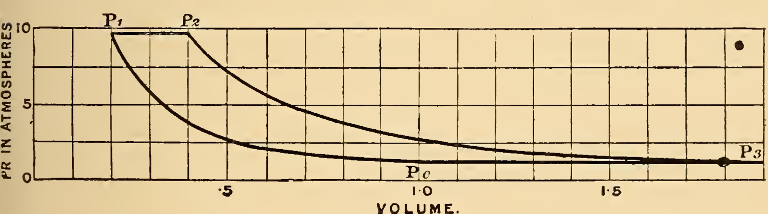




Fig. 5.

No gas engine works on this cycle or indeed on anything very like it. It is not therefore quoted nearly so often in gas engine work as in steam engine practice.

**17. The Constant Pressure Type.** — In this type of engine all the heat is received at the highest *pressure* and rejected at the lowest *pressure*.

Entropy and $P.V.$ diagrams are shown for this cycle in Fig. 5, and it is easily seen from them what the cycle of operations really is. The heat received is clearly $(T_2 - T_1) \times C_p$, where as usual $C_p$ means the specific heat at constant pressure, and that rejected $(T_3 - T_0) \times C_p$, so that the efficiency equals

$$\frac{(T_2-T_1)C_p-(T_3-T_0)C_p}{(T_2-T_1)C_p}=\frac{T_2-T_1-T_3+T_0}{T_2-T_1}.$$

Now when the engine is neither taking in heat nor rejecting it, it must be working adiabatically, i.e. $pV^\gamma$=constant.

Therefore $$p_1V_1^\gamma=p_0V_0^\gamma \text{ and } \frac{p_1V_1}{T_1}=\frac{p_0V_0}{T_0}$$

or $$\frac{T_1}{T_0}=\frac{p_1V_1}{p_0V_0}=\frac{p_1}{p_0}\cdot\left(\frac{p_0}{p_1}\right)^{\frac{1}{\gamma}}=\left(\frac{p_1}{p_0}\right)^{\frac{\gamma-1}{\gamma}}.$$

similarly $$\frac{T_2}{T_3}=\left(\frac{p_1}{p_0}\right)^{\frac{\gamma-1}{\gamma}}$$

Then efficiency $$=\eta=\frac{T_2-T_1-T_3+T_0}{T_2-T_1}=1-\frac{T_3-T_0}{T_2-T_1}$$

and $$\frac{T_1}{T_0}=\frac{T_2}{T_3} \text{ so that } \eta=1-\frac{\frac{T_0}{T_1}\cdot T_2-T_0}{T_2-T_1}$$

or $$\eta=1-T_0\cdot\frac{\frac{T_2}{T_1}-1}{T_2-T_1}$$

$$=1-\frac{T_0}{T_1}$$

$$\eta=1-\left(\frac{p_0}{p_1}\right)^{\frac{\gamma-1}{\gamma}}.$$

Now $V_0$ is the volume at the beginning of compression and $V_1$ at the end of compression, therefore the compression ratio

$$r=\frac{V_0}{V_1} \text{ and } \frac{V_0}{V_1}=\left(\frac{p_1}{p_0}\right)^{\frac{1}{\gamma}}$$

Therefore $$\eta=1-\left(\frac{1}{r^\gamma}\right)^{\frac{\gamma-1}{\gamma}}=1-\left(\frac{1}{r}\right)^{\gamma-1}$$

and this gives the value of the efficiency of this cycle in terms of $r$, the compression ratio. It is an interesting and important fact that this efficiency is independent of the temperatures and pressures attained, and depends only on *the ratio of compression*, i.e. on the relative sizes of the volumes before and after compression. It shows too that for high effi-

ciencies the compression must be high. It must be noted that $T_1$ and $T_0$ do not mean the same thing in this cycle as they did in the Carnot Cycle already referred to. The careful reader will have noted this already. The Brayton and Diesel engines approach most nearly to this cycle.

**18. The Constant Volume Type.**—In this type all the heat is received at constant volume and rejected also at constant volume. These two volumes are the volume at ignition and the volume at exhaust. This cycle may also be called the Otto or Beau de Rochas Cycle, and it is the one on which practically all modern gas engines work or attempt to work. The diagrams in Fig. 6 show the working of the cycle.

The efficiency is calculated in the same manner as the

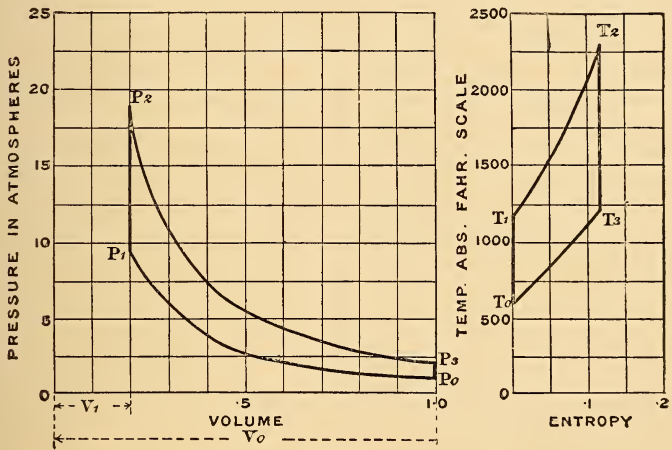



FIG. 6.

previous one; heat taken in $=(T_2-T_1)C_v$ and heat rejected $=(T_3-T_0)C_v$, where as usual $C_v$ means the specific heat at constant volume.

$$\text{Efficiency}=\eta=\frac{(T_2-T_1)C_v-(T_3-T_0)C_v}{(T_2-T_1)C_v}$$

$$\therefore\ \eta=\frac{T_2-T_1-T_3+T_0}{T_2-T_1}$$

$$=1-\frac{T_3-T_0}{T_2-T_1}$$

Then as before

$$\frac{T_1}{T_0}=\frac{T_2}{T_3}=\left(\frac{V_0}{V_1}\right)^{\gamma-1}=r^{\gamma-1}$$

Therefore $$\eta=1-\left(\frac{1}{r}\right)^{\gamma-1}$$

And this it will be noted is exactly the same expression as before. Indeed, the Carnot Cycle can also have its efficiency expressed in exactly the same way, but it must be remembered that although the efficiency of all three cycles depends upon the degree of compression and would be the same in all were the compression ratios the same, yet the temperature ranges would be very different, and it would be found that the Carnot Cycle gave **the least range of temperature for any given efficiency.** The discovery that for the same compression ratios the same efficiency holds good for each of these three cycles is attributed to Professors Unwin and Callendar.

In view of the simplicity of this result it is not difficult to understand that the Committee of the Institution of Civil Engineers, appointed to enquire into the matter, should have selected for use as the best expression for the ideal efficiency the form—

$$\eta=1-\left(\frac{1}{r}\right)^{\gamma-1}$$

This expression therefore holds the place in gas engine work which in the steam engine is filled by the well-known

$$\frac{T_1-T_0}{T_1}.$$

**19.** The remaining point to be considered is **the value to give to** $\gamma$. The gaseous mixture which works in gas engines depends upon whether lighting gas, producer gas, blast furnace gas or coke-oven gas is being employed, and with oil engines yet different mixtures occur. It is evidently impossible therefore to get a value for $\gamma$ which will accurately suit all engines. It must be remembered, however, that the working fluid always consists chiefly of air, and it has been considered by one school of thought that, having

regard to the preponderance of that familiar mixture of oxygen and nitrogen in all internal combustion engines, little error could arise if it were all assumed to be air. The "**Air Standard**" for efficiency resulted. It assumes that air is the working fluid (and that the relatively small quantity of gas is merely used to heat this air by combustion), and that $\gamma$ has the air value of 1·40, so that

$$\eta=1-\left(\frac{1}{r}\right)^{0\cdot4}$$

This expression gives for different values of $r$ the following theoretical efficiencies—

| $r$ | $\eta$ |
|---|---|
| 2 | 0·242 |
| 3 | 0·356 |
| 4 | 0·426 |
| 5 | 0·475 |
| 7 | 0·541 |
| 10 | 0·602 |
| 20 | 0·698 |
| 100 | 0·841 |

In practice 50 to 60 per cent. of these efficiencies are usually obtained, and it is clear that a comparison between different engines can be made by noting what percentage of the ideal efficiency is obtained, in each case, for the compression ratio at which each works. A natural result of this rise of efficiency with compression is that for many years past there has been a movement among engine designers in favour of higher compression pressures. It is this movement which is the chief cause of the great advances that have been made in the heat economy of gas engines. Thus in 1880 a compression pressure of 30 or 40 lb. per sq. inch was usual. Now the compression pressure sometimes goes up to 170 lb. per sq. inch when working with producer gas and with the Diesel oil engine as high as 500 lb. per sq. inch. The effect of high compression pressures is illustrated in practice by the following figures. According to a recent

statement * an engine working with a compression pressure of 120 lb. used 11,500 h.t.u. per h.h.p.-hour, whereas one working with a corresponding pressure of 170 lb. used only 9,500 h.t.u.

**20.** The Council of the Institution of Civil Engineers have very kindly allowed the reproduction of the diagrams in Figs. 4, 5 and 6 from the Final Report † of the Committee on the Efficiency of Internal-Combustion Engines. They were also good enough to permit of the curve in Fig. 7 being reproduced. It shows the heat contents for 1 lb. of air,

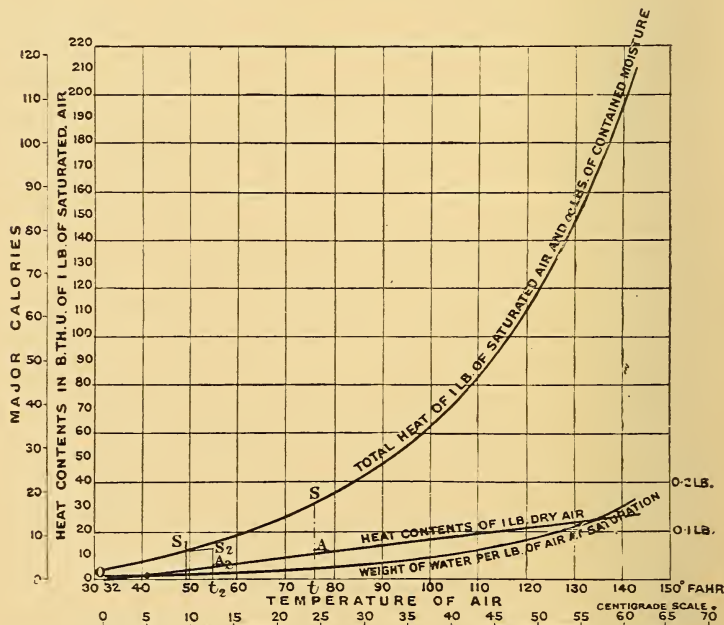



FIG. 7.—Curves showing heat contained in 1 lb. of saturated air at various temperatures. Thus *tS* represents the heat content of 1 lb. of dry air and the associated quantity of water vapour, which occupies the same volume as the 1 lb. of air at temperature *t*.

and the associated quantity of water vapour. It therefore enables the observer to read off at once the heat contained

* Mr. A. E. Porte in *Proc. I.E.E.*, 1907.
† *I.C.E. Proc.*, Vols. 162 and 163.

in any weight of air at different temperatures. The ability to do this rapidly is very useful when a heat balance sheet is being made out for a gas engine run, and reference will be made to it later.

## EXAMPLES.

1. State the laws of perfect gases. Explain what is meant by (1) absolute temperature, (2) a perfect gas, and prove that in a perfect gas $\frac{P.V}{T}$ is constant. A quantity of gas occupying $6\frac{1}{2}$ cubic ft. at temperature 60° F. is compressed isothermally to $\frac{1}{3}$ of its volume. It is then cooled at constant pressure. Find the volume of the gas when the temperature has been lowered in this way to 32° F. *Ans.* 2·05 cu. ft. (Cambridge B.A., 1904.)

2. A vessel is exhausted of air to a pressure of 12 lb. absolute, the pressure of the atmosphere being 15 lb. absolute. The temperature of the whole being that of the atmosphere (60° F.), a cock is opened and air allowed to rush in until the pressure is equalized. Assuming that no heat is lost to the walls of the vessel, find the rise of temperature of the air within it. *Ans.* 130° F. (Mech. Sc. Tripos, Part I, 1905.)

3. Air expands under a piston from a volume of 1 cubic foot and pressure 300 lb. per sq. inch absolute to volume 5 cubic ft. and pressure 40 lb. per sq. inch absolute. Assuming that the pressure and volume vary during the expansion according to the law $PV^n$=const., find the heat absorbed in the process in British Thermal Units. Mechanical equivalent of heat=778 foot-pounds: ratio of specific heats of air=1·41. *Ans.* 74 B.Th.U. (Mech. Sc. Tripos, Part I, 1904.)

4. The entropy of 1 lb. of water at $\theta$° C. is

$$\log_\epsilon \frac{273+\theta}{273}.$$

What is this if $\theta$ is 160° ? If this water is converted into dry saturated steam at 160° C., what is the additional entropy ? *Ans.* 0·461 and 1·14. (B. of E., 1907.)

5. Sketch the compression, ignition, and expansion parts

of a gas engine diagram. If the volumes and pressures at four points on the diagram, to any scales whatsoever, are represented by—

| Points . . . | *A* | *B* | *C* | *D* |
|---|---|---|---|---|
| Volumes . . | 6 | 1·7 | 2 | 4·5 |
| Pressures . . | 1 | 5 | 13·8 | 3·2 |

and if at the point *A* we know that the temperature is 140° C., what are the temperatures at the other points ? Tabulate your results. *Ans.* 140° C ; 313° C ; 1,630° C ; and 720° C.

6. In a gas engine cylinder where $v=2{\cdot}2$ and $p=14{\cdot}72$ it was known that the temperature was 130° C. What is the temperature when $p=122$ and $v=1{\cdot}2$ ? *Ans.* 1,552° C.

7. What is the law connecting the pressure, volume and absolute temperature of 1 lb. of air ? Consult the printed table furnished you, for the density of air. Why is the specific heat greater at constant pressure than at constant volume ? (B. of E., 1900.)

8. What is the law connecting the pressure, volume and absolute temperature of 1 lb. of air ? 1 lb. of air at two atmospheres pressure and 20° C., what is its volume ? It receives heat energy equivalent to 1,000 foot-pounds, its volume remaining constant ; find its new pressure and temperature. The specific heat of air at constant pressure is 0·238. *Ans.* 66·75 cu. ft., 1·15 atms., and 26° C. (B. of E., 1900.)

9. A gas engine works on an ideal cycle with adiabatic compression and expansion, receiving and rejecting heat only at constant volume. Obtain the expression of its efficiency. In such an engine the piston displacement per stroke is 1 cubic foot, the clearance volume 0·2 cubic foot, and at the beginning of compression the temperature of the cylinder contents is 600 F. abs., pressure being atmospheric. The engine receives 0·06 cubic foot of gas per cycle (calorific value 600 B.Th.U. per cubic foot). Atmospheric pressure $=14{\cdot}7$ lb. per sq. in.

Find :—(*a*) Weight of cylinder contents.

(*b*) Pressure and temperature at end of compression (take $\gamma = 1\cdot38$).

(*c*) Rise of temperature during explosion (neglect jacket loss and take $C_v = 0\cdot18$).

(*d*) Pressure at end of explosion.

(*e*) Temperature and pressure at end of expansion.

(*f*) Efficiency of the cycle. *Ans.* 49·4 per cent.

(*g*) Efficiency of an engine working on a Carnot Cycle between the same highest and lowest temperatures. *Ans.* 56·8 per cent.

(Mech. Sc. Tripos, 1906.)

10. Describe, with sketches, the mode of operation of an internal combustion engine. Explain why, in general, such an engine is more efficient as a heat engine than a steam engine of the same power. State where the various losses of energy occur. A gas engine of 10 brake horse-power consumes 180 cu. ft. of gas per hour, the calorific value of which is 690 British Thermal Units per cubic foot. Find its total efficiency, and give a rough estimate of the different proportions of energy lost due to the causes referred to above.

(Mech. Sc. Tripos, Part II, 1906.)

11. A pound of air at atmospheric pressure and 20° C. is to be compressed adiabatically to 10 atmospheres; find the work done by the pump. The same result is arrived at by isothermal compression, cooling the air so that it keeps at 20° C., and when the pressure reaches 10 atmospheres it is heated at constant pressure. The specific heats of air are 0·238 and 0·169. State separately the work done upon and by the air and the heat taken from and given to it, all in foot-pounds. *Ans.* Adiabatic compression 65,470 ft.-lb., isothermal heat rejected 64,825 ft.-lb., added at constant pressure 92,170 ft.-lb.

12. Criticise the "Otto" Cycle, in gas engines, from the point of view of (1) efficiency, (2) relation of power to weight on the part of the engine. In modern practice the tendency is to compress the mixture highly before ignition. How does this affect the points of your criticism?

# CHAPTER III

# Combustion and Explosion

Chemical Combustion—Dugald Clerk's and Grover's Early Experiments on Explosion in Closed Vessels—Discussion of Results—Increase of Specific Heats of Gases—Dissociation—"After-burning"—Later Experiments.

**21. Chemical Combustion.**—Instances of chemical combustion are manifold. Two among the commonest are the burning of coal, and the oxidation of the carbon in food which is the source of the heat energy given out by the human body. In place of coal, it is possible to burn gas made from coal and so obtain either heat or light. In a gas engine cylinder gas and air are first mixed together and the whole mass ignited at once, so that the union is explosive. Useful figures to remember are that 1 lb. of coal on being burnt will liberate about 12,000,000 ft.-lb. of energy, a cubic foot of coal gas will liberate about 550,000 ft.-lb., 1 lb. of petroleum about 18,000,000 ft.-lb., and 1 lb. of petrol some 15,000,000 ft.-lb. These are very large amounts, and were it possible to invent a heat engine of 100 per cent. efficiency it is plain that a very liberal supply of energy would be obtainable at little cost. With existing engines 1 lb. of coal with potential energy equal to 12,000,000 ft.-lb. will only give in energy on the brake about 2,000,000 *ft.-lb. with the best steam engines* and 4,000,000 *ft.-lb. with the best gas engines*, the waste energy being 10,000,000 ft.-lb. and 8,000,000 ft.-lb. respectively in the two cases.

The loss of 8,000,000 ft.-lb. which occurs in a gas engine is divided between the loss to the water in the cooling jacket and the loss which occurs owing to the exhaust

products being at a high temperature and so carrying off a large unutilized portion of the heat. The loss to the water jacket is the more difficult to follow in all the intricacies of the working cycle. The cooling jacket is necessary, as without it the piston and cylinder would get almost red hot and the engine would stop running. How the temperature-flow through the metal depends on the position of the piston in its stroke, is difficult to determine.

**22.** If after a charge of gas and air has been drawn into a gas engine cylinder the flywheel be held so that it cannot move and the charge be then ignited, a rapid rise of pressure is recorded on the indicator. It ought, one would think, to be easy to calculate what this rise would be, since the quantity of gas and air admitted and their quality are easily determinable and the amount of thermal energy liberated therefore known. If this amount of energy be divided by the amount of heat required to heat the mixture through one degree Cent. it is clear that the resulting temperature would be ascertained, and from this it would be simple by the $\frac{PV}{T}$ law to determine the resulting pressure. This has often been done, but it has always been found that the pressure actually obtained is only **about one-half** that calculated. Here are the actual figures obtained in some experiments carried out by Mr. Dugald Clerk—

| Ratio air/gas. | Absolute Pressure Obtained. | Absolute Pressure Calculated. |
|---|---|---|
| 14 | 55 | 110 |
| 13 | $66\frac{1}{2}$ | 116 |
| 12 | 75 | 123 |
| 11 | 76 | 132 |
| 9 | 93 | 161 |
| 7 | 102 | 190 |
| 6 | 105 | 214 |
| 5 | 106 | 206 |
| 4 | 95 | 196 |

On an average there appears here to be a loss of as much as 50 per cent. of the pressure. Why is this ?

**23.** Several **explanations** have been put forward to account for this loss. The most important are—

1. *The Dissociation Theory.*—It is well known that chemical compounds such as $H_2O$ or $CO_2$ dissociate at high temperatures into simpler gases and in so doing absorb heat. It has therefore been thought that at the high temperatures of explosion such dissociation would occur and the heat so absorbed might account for the missing 50 per cent. This assumption, however, involves the deduction that for weak explosions, in which low pressures and temperatures were attained the effect should be much less, so that the actual pressure would form a much larger proportion of the calculated pressure and the converse in the case of rich mixtures. As a glance at the above figures will show, this, however, is not the case; at the weakest mixture of 1 to 14 the missing pressure is 50 per cent., and at the richest of 4 to 1 it is 52 per cent., or practically the same. This theory therefore does not suffice alone to account for the observed facts.

2. *The Cooling Theory.*—This assumes that the cooling effect of the cylinder walls is so great that the pressure actually obtained must fall much below the ideal calculated. It does not explain, however, why the loss should be always 50 per cent. in the particular cylinder used, nor, moreover, does it explain why a 50 per cent. loss is found still to occur even when a cylinder of a different size and shape is chosen. So that this theory also is inadequate in itself to explain the observed effect.

3. *The Increasing Specific Heat Theory.*—This is the theory advanced first by MM. Mallard and Le Chatelier who found as the result of their experiments that the specific heat of gases and particularly of $CO_2$ appeared to increase considerably with rise of temperature. The objection commonly alleged against this theory is that, as in the *Dissociation Theory*, it requires that a greater proportion of the ideal pressure should be obtained at lower temperatures than at higher, and that this is not found to be the case.

4. *The After-burning Theory.*—This theory has chiefly

been associated with the name of Mr. Dugald Clerk, who suggested that the combustion of the gas was not as rapid as supposed and that not all the heat was liberated before the moment of highest pressure. It assumes in fact that the gas is still burning long after the point of maximum pressure and that the cooling effect of the walls has therefore a much longer time to operate than had been generally supposed. In an actual gas engine this would mean that the gas would be burning right through the working stroke and that it must sometimes happen that unburnt gas would pass away in the exhaust. The objection to this theory lies in the fact that it has never been shown conclusively that the explosion is not complete at the point of highest temperature. Indeed the evidence is rather the other way. It is not usual to find the exhaust to contain more than a very few per cent. of unburnt gases, and Prof. Burstall has shown that a complete heat balance analysis can be obtained without the need of any such hypothesis.

**24.** Thus there are **four** simple theories, none of which appear to be sufficient in themselves to account for the observed loss. The difficulty is so fundamental a one that still further theories compounded of the above have been put forward. Mr. Dugald Clerk has recently suggested, as will be explained later at greater length, that the supposed 50 per cent. may be accounted for on the supposition that part is due to the after-burning loss and part to a certain increase in specific heats. The author has seen no reason to alter the suggestion he himself made at the meeting of the British Association * in 1902, viz. that the so-called "suppression of temperature" was probably due to the combined action of cooling and of increase of specific heat on the lines suggested by the French physicists, MM. Mallard and Le Chatelier. It was shown that although the increase of specific heat left a larger proportion of loss to be accounted for at low temperatures than at high ones, this was sufficiently explained by the fact that the ignition period was much longer at low temperatures and so

* *The Engineer*, October 10, 1902 ; *Engineering*, October 10, 1902.

allowed the cooling effect to have a longer time for action than it would have at high temperatures. This meant that for weak mixtures the 50 per cent. loss was mainly due to cooling, for rich mixtures mainly due to increase of specific heat, and for intermediate mixtures was due to a combination of the two.

**Mr. Dugald Clerk's early experiments** had consisted in indicating explosions of mixtures of air with Glasgow and Oldham gas in a closed cylinder 7 in. by $8\frac{1}{4}$ in. The indicator registered pressure $p$ on a rotating drum driven at a known constant speed, so that curves were obtained showing the relation between $p$ (pressure) and $t$ (time) during the explosion and the subsequent cooling of the gas to the walls and ends of the cylinder. From the diagrams so obtained it was of course possible for the author to measure the time occupied by the explosion, and the subsequent rate of fall of pressure due to cooling. The specific heat constants were taken from the experiments of MM. Mallard and Le Chatelier as reduced by Prof. Burstall in his report of the Gas Engine Research Committee of the Institution of Mechanical Engineers. That there are objections to the method of experimenting by which the French physicists obtained their results is well known. In fact Prof. Callendar has remarked: "The method of experiment employed was closely analogous to the explosion that was taking place in the gas engine itself. Explosive mixtures were fired in a closed cylinder 17 in. by 7 in., and the maximum pressure was read by means of a Bourdon gauge." Since the date of this paper other measurements have been made, and though the results obtained vary among themselves it may be said that on the whole they support the general conclusion reached by MM. Mallard and Le Chatelier. If the theoretical temperature of explosion is calculated from these values of the specific heat the difference from the observed value is much less. Thus a column may now be added to the table last given—

| Ratio air/gas. | Absolute Pressure Obtained. | Absolute Pressure Calculated on Constant Specific Heat. | Absolute Pressure Calculated on Variable Specific Heat. |
|---|---|---|---|
| 14 | 55 | 110 | 83 |
| 13 | 66½ | 116 | 86 |
| 12 | 75 | 123 | 90½ |
| 11 | 76 | 132 | 95 |
| 9 | 93 | 161 | 107 |
| 7 | 102 | 190 | 121 |
| 6 | 105 | 214 | 131 |
| 5 | 106 | 206 | 127 |
| 4 | 95 | 196 | 123 |

It will be seen that in the case of the weakest mixture the 50 per cent. loss has been reduced to 34 per cent., and in the case of the richest 52 per cent. has been reduced to 23 per cent., showing a step in the required direction. It is now necessary to make some allowance for the cooling and see how far the discrepancy still remaining may be accounted for. The basis on which such an allowance can be made is as follows : the law connecting $p$ and $t$ (in seconds) during cooling is ascertainable from the curves given by the indicator, and it is therefore possible to calculate what the rate of loss of heat energy would be at the actual maximum temperature of explosion. This rate is of course greater than the mean rate from the beginning of explosion, since initially the temperature of the gas was approximately atmospheric. Starting, however, as it does from zero and rising quickly to a definite maximum the mean value for the very brief interval of explosion may, to a first approximation, be taken as half the maximum.

In view of the very considerable discrepancy to be accounted for—in all some 50 per cent. of the total energy—small variations in the constants adopted would be of relatively little moment. In the absence, therefore, of other data the writer adopted those for lighting gas as given in the I.M.E. Report. Change of volume due to chemical combustion being of very small effect is neglected. From the cooling curve it was found that $p=\frac{31}{\sqrt{t}}$, and from this

it can be calculated that the losses due to cooling must be such as to give the final temperatures included in the following completed table—

| Air. / Gas. | Maximum Pressure on Explosion. lb./in.². | | | |
|---|---|---|---|---|
| | On Constant Specific Heat Hyp. | On Variable Specific Heat Hyp. | Variable Specific Heat with Cooling Allowance. | As obtained by Mr. Dugald Clerk on experiment. |
| 14 | 110 | 83 | 60 | 55 |
| 13 | 116 | 86 | 66 | 66 |
| 12 | 123 | 90 | 70 | 75 |
| 11 | 132 | 95 | 76 | 76 |
| 9 | 161 | 107 | 91 | 93 |
| 7 | 190 | 121 | 104 | 102 |
| 6 | 214 | 131 | 115 | 105 |
| 5 | 206 | 127 | 109 | 106 |
| 4 | 196 | 123 | 93 | 95 |

The results are also shown as a curve in Fig. 8, and it will be admitted that the pressures so calculated in this way agree very well with those obtained in the experiments. This expression for the rate of cooling of gaseous mixtures enclosed in metal cylinders of stated dimensions is not easy to apply to the case of ordinary gas engines. First, because the connexion between the rate of loss of heat and the dimensions of the cylinder is very complicated, but even more because in an ordinary gas engine cylinder the temperature of the cylinder walls and of the piston * are so very different that conditions sometimes arise in which while the gas is being heated by the piston it is at the same time being cooled by the cylinder walls, a condition of affairs in no way analogous to that holding in the above experiments.

**25.** At the time these calculations were made the only other well-known experiments upon the explosion of gases in closed vessels were those of Mr. Grover, and at the British Association meeting in 1903 the author presented a paper in which an endeavour was made to show how far the combined variable specific heat and cooling theory would go towards explaining the very remarkable results obtained

* See *Engineering*, June 27, 1902, and August 1, 1902.

by Mr. Grover, which in no way resembled those obtained by Mr. Dugald Clerk, inasmuch as the former found much lower pressures and came to the unexpected conclusion that the retention of waste products in a gas engine cylinder increased the pressure of the ensuing explosion, an astonishing result having regard to the great care

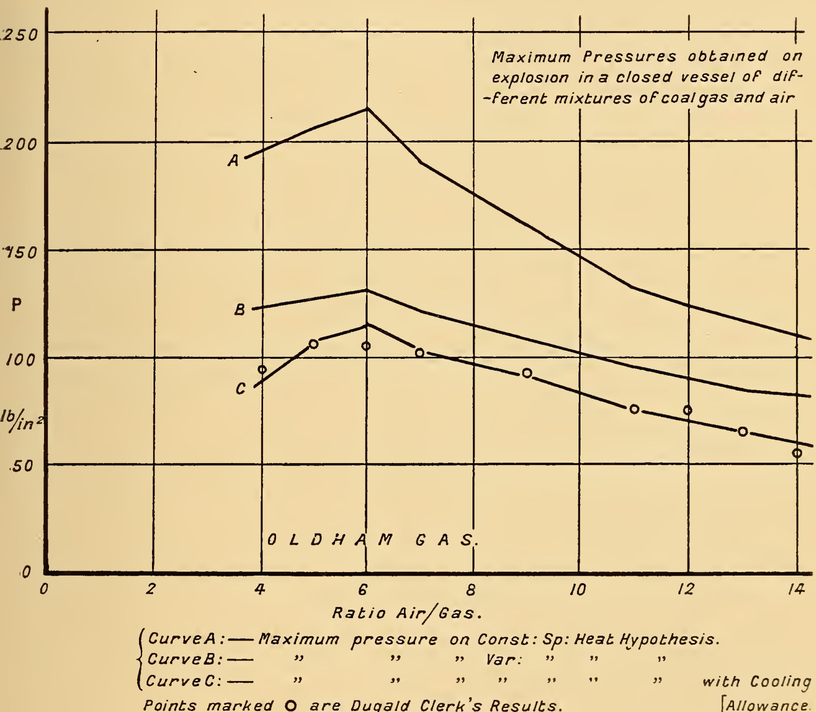



FIG. 8.

taken by most gas engine manufacturers to sweep out the greatest possible amount of the products of old explosions. The great difference between the maximum pressures obtained by Mr. Dugald Clerk and Mr. Grover is illustrated in the figure that follows.*

It was Mr. Grover's idea not only to measure the pres-

* See also *Engineering*, September 18, 1903.

sures produced by various richnesses of mixture of coal-gas and air, but to investigate whether the resultant pressure on explosion was affected by replacing the air in excess of that calculated as chemically necessary for complete combustion by a portion of the burnt products of the previous explosion. It appears from Mr. Grover's account of the experiments that he had an iron cylinder of one cubic foot capacity, and that in each series of experiments the volume of the coal gas admitted was kept constant and the cylinder was then filled with a mixture of air and waste products in various proportions. This was done in each series by filling the cylinder with water, and allowing gas to enter whilst a known volume of water was run out. Thus after an explosion, water was allowed to pass into the cylinder until all but the required volume of burnt products had been forced out ; so that if it were desired that no burnt products should be left, the cylinder would be completely filled with water, but if, say, 50 per cent. of the volume of the cylinder was required to contain burnt products, the water would only be permitted to rise half-way up the cylinder.

The pressure was recorded in the customary manner on a rotating drum, but very few of the curves are given in the published account of the experiments, and it is therefore difficult to make a very exact comparison between the time rate of fall of the pressure after explosion in Mr. Grover's experiments (using, of course, those experiments in which no burnt products were admitted) with those of Mr. Dugald Clerk. However, so far as the curves can be examined, they show for the same pressures almost exactly the same rate of fall, a result which is the less unexpected, as the diameters of the two cylinders appear to have been nearly equal. It is not difficult to calculate what the ideal maximum temperature and corresponding pressure of explosion would be were there a variable specific heat, but no cooling of the gas by the walls, and when this has been done, it may be compared with the pressure found experimentally. The following table shows the result of such a calculation—

TABLE I.

| Ratio of Gas to Air. | Pressure Observed. | Pressure Calculated. | Corresponding Difference in Energy (Approx.). |
|---|---|---|---|
| | (Absol.) | (Absol.) | Ft.-lb. |
| 15 | 31 | 73 | 21,000 |
| 14 | 39 | 76 | 19,000 |
| 13 | 46 | 79 | 18,000 |
| 12 | 51 | 83 | 18,000 |
| 10 | 63 | 92 | 18,000 |
| 8 | 77 | 104 | 18,000 |
| 6 | 77 | 119 | 31,000 |

In this table there is also given the difference in the

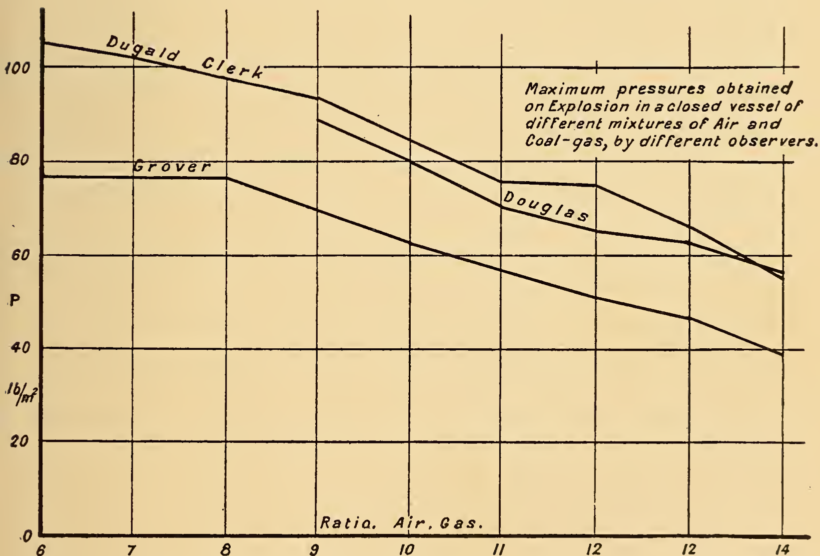



FIG. 9.—Explosion curves showing much lower pressures obtained by Mr. Grover than by other observers.

heat energy between the gas at this temperature and at the actual temperature attained. These amounts were found in Mr. Dugald Clerk's experiments to be of almost exactly the calculated amount that would be expected to be lost to the walls and ends of the cylinder by cooling during the time of explosion. Such a hypothesis, however, does

not fit Mr. Grover's results. Nor is it to be expected—having regard to the extreme divergence between their results—that both would be susceptible to the same treatment.

The above curves show the actual pressures plotted with respect to richness of mixture for the experiments of both investigators. It is seen that Mr. Grover's curve lies far below Mr. Dugald Clerk's. This cannot be due entirely to the different cylinder volumes used (317 and 1,728 cubic inches), or to differences in the chemical constitution of the gases, because, as will be seen from th eintermediate curve, there is little disagreement between the results obtained by Mr. Dugald Clerk and Mr. Douglas, although the results * obtained by the latter were for gases enclosed, not in iron cylinders, but in a eudiometer tube. If the use of a eudiometer does not produce results more different from Mr. Dugald Clerk's than this, the presumption certainly is that some factor must have entered into Mr. Grover's experiments which has entirely masked his results. A suggestion as to what this factor could be has been made by Mr. Grover himself, for in describing one experiment he says : "The difference is no doubt due to the fact that water was present on the walls of the cylinder ; " but Mr. Grover did not consider apparently that this presence of water affected his conclusions on the subject generally—conclusions which are set forth on pp. 231 and 232 of his *Modern Gas and Oil Engines*. In his paper the author had no hesitation in attributing not only the discrepancy in the heat balance-sheet of the experiments, but also the extraordinary results obtained in the allied series of experiments, in which burnt products were present, to the effect of a water film.

There is, of course, a limit to the quantity of water which could adhere to the walls of the cylinder, and it is necessary to see whether the required amount is what could reasonably be expected to exist. The average loss of energy given in column 4 of the table on p. 45 is about

* See *The Engineer*, April 22, 1887, and November 7, 1902.

20,000 foot-pounds, and considering the average energy given to 1 lb. of water to raise it from atmospheric temperature to superheated steam at the average maximum temperature, as about 750 Cent. heat units, it follows that the weight of water required equals 0·0191 lb. This would occupy a space of about 0·53 cubic inch, and in a cylinder of the dimensions used, a film of water $\frac{1}{2000}$ in. thick would be sufficient to account for this. So that there is no difficulty in accounting for the presence of a sufficient quantity of water. It is interesting to assume the presence of this small quantity of water, and to trace its effect throughout the whole explosion. As it happens, the actual calculations involved are a little tedious; as the temperature rises the vapour tension of the water increases until a time comes when steam is given off; this, of course, does not occur at the ordinary boiling point, as by this time the gases in the cylinder will be at a pressure in excess of the atmospheric pressure. The temperature still rises, but so long as there is any water left, the steam is saturated and the temperature of the gases must therefore keep step with the pressure—that is, if the contents of the cylinder are assumed to be all at the same temperature at the same time, and without such an assumption calculation is out of the question. Next, a point is reached at which the steam begins to be superheated; further, the formation of a volume of steam has meant the compression of the gases in the cylinder so altering the relation between the pressure and the temperature, and bringing a further complication into the matter. It therefore becomes a matter for careful treatment to draw out a temperature entropy curve for the whole. (Perhaps the simplest way is to think of the gas and the water stuff being kept separated by a thin diaphragm, but kept at the same temperature at the same time.) To show the result of the steam compressing the gas a simple example may be taken, in which the weight of the film of water is three-eighths of the weight of the gaseous mixture, and in which both start at atmospheric temperature and pressure. When the pressure (absolute)

amounted to 33 lb. per square inch, a calculation made in the absence of the knowledge of the presence of a water film would give a temperature of 254° Cent., whereas the real temperature would be 124° Cent., a very different result. A further calculation with the same amount of water present shows that a pressure of 60 lb. per square inch would be attained on explosion, whereas under the same circumstances, but in the absence of the water film, a pressure of 100 lb. per square inch would have been attained.

The curves and tables above given are for the experiments made by Mr. Grover with mixtures of coal-gas and air only. No burnt products were present. When burnt products were admitted, very remarkable results were observed. Taking the first series * of experiments (in which the volume of coal-gas was one-sixteenth of the total volume of the cylinder), when there were no burnt products present, the pressure recorded was 16 lb. per square inch above the atmosphere. When burnt products were present to the extent of 1/3·3 of the volume of the cylinder the pressure rose to 35 lb. per square inch above the atmosphere.

Now it will be observed that in the experiment in which burnt products were absent, the whole of the interior of the cylinder must have been wetted by the water used in the measurement of the volumes, and on the other hand, when burnt products were present, the water only rose up to two-thirds of the height of the cylinder. Hence, in the second experiment, one-third of the exposed surface was dry, and would therefore cool the gases in the manner already observed in Mr. Dugald Clerk's experiments, whilst the remaining two-thirds of the surface was covered by a water film which would, as explained above, absorb much of the heat energy of the gas before it was evaporated. When the whole surface was wet, only 16 lb. per square inch was registered; and when one-third was dry and two-thirds wet the pressure rose to 35 lb. per square inch.

The author concluded that the presence of a water film of varying extent was a sufficient explanation of the very

* Grover's *Modern Gas and Oil Engines*, 1902 edition, p. 233.

curious results obtained by Mr. Grover. The hypothesis upon which this explanation was made leads, however, to a further deduction, the truth of which it remained to investigate. Mr. Dugald Clerk's curves showed that the rate of loss of heat energy to the walls increased much more rapidly than the temperature of the gas; in fact, the rate of loss was about proportional to the third power of the absolute temperature. It followed, therefore, that in the richer mixtures in which higher temperatures would be attained, the increase in the loss owing to the increased cooling effect of the fraction of dry wall exposed, would be much greater than the saving due to the water film only covering two-thirds instead of the whole of the surface, and that in consequence, for the richer mixtures, the apparent effect of burnt products in increasing the resultant pressure would be much diminished, if not extinguished altogether. Now this was precisely what had been found by Mr. Grover to be the case. When the ratio of volume of gas to volume of cylinder was 1 to 13, the effect of burnt products was to increase the pressure from 36 lb. to 43 lb. per square inch—a far smaller increase than before; and when the ratio was increased to 1 to 9, the effect was quite wiped out, and the pressure fell almost immediately the burnt products were admitted. It is not, however, surprising that the unexpected results found by Mr. Grover should be capable of being accounted for without it being necessary to assume that the burnt products really would increase the pressure under such conditions as usually hold in a gas engine cylinder. The only reason why burnt products might exercise any such tendency would lie in their having a smaller specific heat than the explosive mixture, which is not the case. In practice it has been found that the presence of burnt products in the explosive mixture has anything but a good influence on the economy of gas engines.

The results of this investigation into these early experiments on gaseous explosion, made it appear to the author that it could not be wise, in gas engine calculations, to assume the constancy of the specific heats of the working gases. Indeed, that it might be greatly doubted whether such

calculations could be regarded as of permanent value, unless an alternative calculation founded on the basis of a variable specific heat was also given. In his calculations for the Institution of Mechanical Engineers, a linear equation connecting specific heat with temperature was used by Professor Burstall,* and it was found to lead to consistent results.

26. *Exercise.*—What percentage of dissociation of $CO_2$ to CO and O at, say, 1,700° C. would suffice to mask the effect of the rise of true specific heat ? To solve this it must be remembered that if we have 44 kg. of $CO_2$ and 1 per cent. of it is dissociated to CO and O it would require an absorption of heat equal to 682 calories (*see* Chapter VI. as to this) ; also that the rise in true specific heat would probably be of the order 0·2, corresponding to an additional absorption of heat of $\frac{1700}{2} \times 0{\cdot}2$ calories per kg. of $CO_2$ or $\frac{1700}{2} \times 8{\cdot}8$ calories for 44 kg. which comes to 7,480 calories.

It follows from this that 1 per cent. of dissociation would cause the apparent specific heat to come out at about 11 per cent. higher than the value of the true specific heat. In connexion with this it may be remembered that Langen could detect no evidence of dissociation up to 1,700° C. in his explosion experiments. On the other hand it is clear that 1 per cent. of dissociation could materially affect specific heat measurements and that 10 per cent. would be near to doubling the apparent rate of increase of specific heat.

**27. Later Experiments.**—So much for the earlier experiments on the combustion of gas and air ; the writer has stated how a number of years ago he came to the conclusion that increase of specific heat was the determining cause in the apparent " suppression of heat," and a further illustration will be given in the following chapter. In addition to the early experiments of Dugald Clerk and Grover, some work in the same direction was done at the Massachusetts Institute of Technology, but it does not appear

* Professor Burstall appears to have made his calculations thus :—The exact amount of energy liberated on explosion was known. The variable specific heat allowed of three-fourths of the energy being accounted for. It was then assumed that the remainder was lost to the water-jacket during explosion, the amount so lost during expansion being taken as the difference between the work done and the change in the internal energy of the gas. The same process being carried out during the compression, the net loss to the jacket per cycle could be calculated. This should agree with the observed loss owing to heating of the water-jacket, and in practice a satisfactory agreement was observed.

that any definite conclusions were drawn therefrom. Later experiments have been made by Mr. Dugald Clerk, Professor Hopkinson and Messrs Bairstow and Alexander. Taking the last first: Messrs. Bairstow and Alexander's experiments were made on mixtures of London coal-gas and air in a cylinder 18 in. long and 10 in. in diameter, pressures were indicated on a rotating drum, and the results of the investigations were communicated to the Southport meeting of the British Association in 1903. The conclusions reached were not, however, in accordance with the still later experimental results obtained by Mr. Dugald Clerk and Professor Hopkinson, and it is not necessary to say more than that Messrs. Bairstow and Alexander considered that "at the high pressure reached by the best explosive mixtures, the loss due to cooling is less than the errors of observation, the calculated and actual values differing only on account of dissociation. At the lower pressures given by the weaker mixtures the whole of the loss of pressure is due to cooling. This shows a temperature of about 1,200° Cent. before carbon dioxide begins to dissociate." It will be seen that these conclusions are different from those which the author arrived at as the result of an investigation of other experimental results. In a paper communicated to the Royal Society two years later the same authors repeat their conclusions in more explicit language :—"Mixtures of coal-gas and air are not inflammable until the volume of coal-gas is greater than one-seventeenth of the combined volumes. Only a very small fraction of the gas then burns, the amount rapidly increasing with increased richness of the mixture until the coal-gas is one-twelfth of the total volume. The least inflammable of the constituents then burns, and combustion becomes and remains complete so long as air is in excess. In these latter cases it is still probable that the constituents burn successively and not simultaneously. The hypothesis of a specific heat increasing with temperature is not supported by direct experiment, and cannot be proved by any work on the pressures produced by explosion, the authors believing that a proof would require the measurement of temperature. Direct experiments by Deville

at temperatures below 1,400° C. have shown that both steam and carbon dioxide are partially decomposed, and this dissociation is therefore taken by us as the sole explanation of the difference between the pressures calculated for explosions in a closed vessel and those actually obtained." As will be seen later, Professor Hopkinson has since made direct measurements of the temperatures.

**28.** In 1906 Mr. Dugald Clerk communicated to the Royal Society the results of some experiments he had made by a new method. It had often been remarked that the expansion curve on gas engine diagrams was found to lie above the adiabatic line, and many deductions had been made therefrom. Mr. Dugald Clerk's new method of experiment consisted in running a gas engine under the ordinary standard conditions and then at a given moment preventing the exhaust and inlet valves from opening, and at the same time taking a series of indicator diagrams. These diagrams showed a number of expansion and compression curves with the pressures gradually falling as the gas cooled. Fig. 10 is a representation of a series of curves so obtained. From the shape

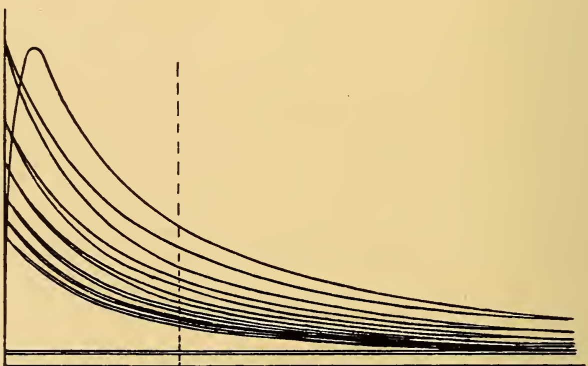


FIG. 10.—Dugald Clerk's "Zig-Zag" curves. 1/9 mixture.

of these curves it was possible to make refined calculations as to what was occurring to the gas in the cylinder. The average temperature must clearly diminish owing to the effect of the water-cooled walls, whilst if combustion continues after the point of maximum temperature an effect

must be observed in an apparent lessening or even reversing of the cooling action. This is merely a general idea of the method of experiment and calculation. In its details it consists of a series of secondary approximations by means of secondary curves, in order to ascertain what the apparent specific heat of the mixture must be at any given temperature and at any point of the stroke. In these calculations the work done by the flywheel on the gas is allowed for. The values of the apparent specific heats at 1,000° C. and 1,500° C. as found by this well-known authority agree fairly well with the values calculated from the Mallard and Le Chatelier formulae, viz.—

For $CO_2$; $C_v = 0{\cdot}1423 + 0{\cdot}0000834T$
$H_2O$; $C_v = 0{\cdot}3116 + 0{\cdot}000182T$
$N_2$; $C_v = 0{\cdot}171 + 0{\cdot}0000215T$
$O_2$; $C_v = 0{\cdot}150 + 0{\cdot}0000188T$

but as regards the "suppression of heat" controversy his conclusions are :

"(1) The apparent specific heat of the working fluid of the internal combustion engine (consisting mainly of nitrogen, carbon dioxide, steam and oxygen), when calculated from the first three-tenths of the engine stroke, undoubtedly increases between the observed temperatures 300° C. and 1,500° C., but tends to a limit at the upper temperature.

"(2) The apparent change in specific heat is not entirely due to a real change in specific heat, but requires in addition continuing combustion to account for all the facts.

"(3) The rate of heat-flow from the working fluid to its enclosing walls for equal temperature differences varies throughout the stroke. Increased heat-flow accompanies increased mean density.

"(4) The mean temperature of the inner surface of the enclosing walls varies with the portion of the stroke examined from 190° C. for whole stroke to 400° C. for first three-tenth stroke under working conditions at full load. These mean temperatures, however, are not the highest mean temperatures reached by the walls.

"(5) The heat distribution during the operation of the working fluid can be determined with approximate accuracy from the apparent specific heat values and heat-flow values obtained from the diagram only."

It will be seen that these conclusions show some modification of the position taken up earlier by Mr. Dugald Clerk, although he still considers that combustion is not complete at the point of highest pressure. To quote his own words: "When the writer began the present investigation he believed that these phenomena of slower chemical action furnished a complete explanation of the discord between the theoretical and observed results, and that there was no need to assume any considerable dissociation or variation of specific heat of the products of combustion. These experiments, however, appear to him to indicate real change of specific heat as well as continuation of combustion. The experiments do not exclude dissociation or any other molecular change which by requiring the performance of work would change specific heat. It appears improbable, however, that dissociation should be material for temperatures so low as 600° C. It is not usual to suppose that either carbon dioxide or steam can be decomposed to any sensible extent at such temperatures."

In giving due weight to the conclusions so reached, it must not be forgotten that the method is essentially one of small differences, and the experimenter himself admits that the whole of his deductions could be greatly affected by an inaccuracy in the indicator curves of one-fiftieth part of an inch and affected not a little by an error of $\frac{1}{100}$ inch. Having regard to the circumstances, in respect of inertia and time-lag, in which even the best indicator works, it will be realized that too much reliance must not be placed upon the results obtained. There is no doubt that Mr. Dugald Clerk used an exceptionally accurate instrument and treated his diagrams with great skill; as a piece of difficult experimental work he is certainly entitled to congratulation on the results achieved. The actual figures obtained by Mr. Dugald Clerk for the specific heat are shown in the following two tables.

TABLE I.—APPARENT SPECIFIC HEATS (INSTANTANEOUS) IN *Foot-pounds per Cubic Foot* OF WORKING FLUID AT 0° C. AND 760 MM.

| Temperature. | Specific Heat at Constant Volume. | Temperature. | Specific Heat at Constant Volume. |
|---|---|---|---|
| Degrees C. | Ft.-lb. | Degrees C. | Ft.-lb. |
| 0 | 19·6 | 800 | 26·2 |
| 100 | 20·9 | 900 | 26·6 |
| 200 | 22·0 | 1,000 | 26·8 |
| 300 | 23·0 | 1,100 | 27·0 |
| 400 | 23·9 | 1,200 | 27·2 |
| 500 | 24·8 | 1,300 | 27·3 |
| 600 | 25·2 | 1,400 | 27·35 |
| 700 | 25·7 | 1,500 | 27·45 |

TABLE II.—MEAN APPARENT SPECIFIC HEATS IN FOOT-POUNDS PER CUBIC FOOT OF WORKING FLUID AT 0° C. AND 760 MM.

| Temperature. | Specific Heat at Constant Volume. | Temperature. | Specific Heat at Constant Volume. |
|---|---|---|---|
| Degrees C. | Ft.-lb. | Degrees C. | Ft.-lb. |
| 0—100 | 20·3 | 0—900 | 23·9 |
| 0—200 | 20·9 | 0—900 | 24·1 |
| 0—300 | 21·4 | 0—1,100 | 24·4 |
| 0—400 | 21·9 | 0—1,200 | 24·6 |
| 0—500 | 22·4 | 0—1,300 | 24·8 |
| 0—600 | 22·8 | 0—1,400 | 25·0 |
| 0—700 | 23·2 | 0—1,500 | 25·2 |
| 0—800 | 23·6 | — | — |

It will be observed from the above that although the apparent specific heat was found to increase with rise of temperature, it tended towards a limiting value. The increase found for the first 500° C. was far more than for the last 500°. This conclusion is not in accord with the experiments of other workers.

**29.** The last series of experiments of this kind to be described are those of Prof. Hopkinson, communicated to the Royal Society in 1906. These were explosion experiments in a closed vessel as shown in Fig. 11. *A* is the sparking point, *B*, *C* and *D* are platinum thermometers.

Thermometer $B$ is practically at the centre of the vessel, $C$ is about 30 cm. distant from the spark and $D$ is about 1 cm. from the walls of the vessel. A record of the pressure was taken on the same drum as that upon which the tem-

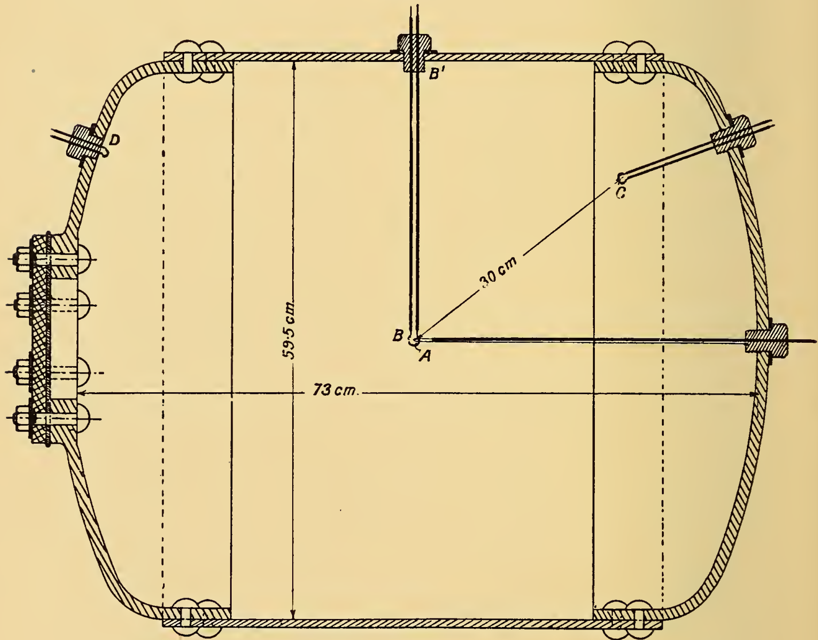



Fig. 11.—Professor Hopkinson's Gas Explosion Apparatus.

peratures were electrically recorded. The indicator was very simple, consisting as it did of a piston controlled by a flat steel spring held at the two ends. As the spring was deflected a mirror tilted and so threw a beam of light on to the moving film. The period of the instrument was about $\frac{1}{300}$ sec. Fig. 12 shows the kind of result obtained when plotted out. The following table serves also to show the actual indications recorded by the electric thermometer placed at the centre of the vessel :—

| Time. Secs. | Resistance. Ohms. | Rise of Resistance. Ohms. | Temperature in Degrees C. |
|---|---|---|---|
| 0·008 | 22·05 | 12·4 | 560 |
| 0·024 | 30·3 | 20·7 | 995 |
| 0·041 | 32·7 | 23·1 | 1,135 |
| 0·057 | 33·1 | 23·5 | 1,165 |
| 0·074 | 33·1 | 23·5 | 1,165 |
| 0·09 | 34·0 | 24·4 | 1,225 |
| 0·107 | 34·5 | 24·9 | 1,260 |
| 0·123 | 34·7 | 25·1 | 1,275 |
| 0·140 | 34·7 | 25·1 | 1,275 |
| 0·173 | 36·6 | 27·0 | 1,400 |
| 0·26 | wire melts | — | 1,710 |

An investigation had also to be made into the question of the existence of a time and temperature lag in the tem-

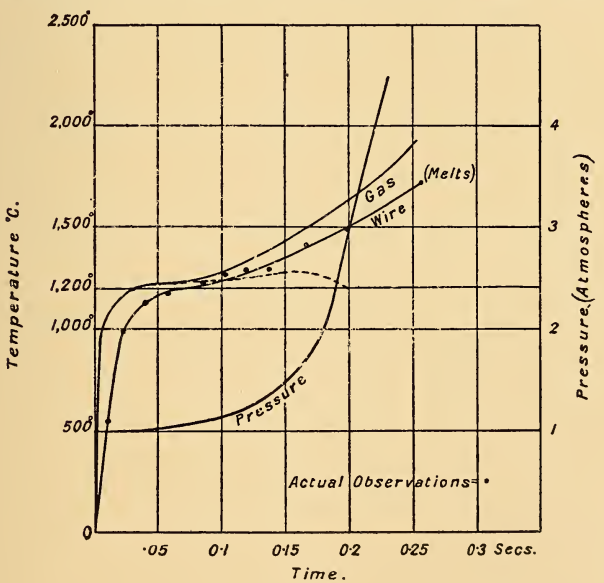



FIG. 12.

perature recorded by the thin platinum wire. A theoretical investigation of this problem was carried out by the writer some years ago when contemplating undertaking a series of experiments similar to those of Prof. Hopkinson, but

which had to be set on one side owing to the pressure of other work. It suffices to say here that Prof. Hopkinson found the temperature of the wire to lag materially behind that of the gas when the latter was changing rapidly. To test this, wires of two different thicknesses were used, viz. $\frac{1}{1000}$ in. and $\frac{1}{2000}$ in. respectively, and by a comparison of the results obtained Prof. Hopkinson was able to find the amount of the correction which he considered it necessary to employ.

The most important of the conclusions reached by this experimenter, who, he tells his readers, carried out these experiments, largely "with the object of finding the cause of the so-called 'suppression of heat' in explosions," is that his experiments appear to prove that *even in the weakest mixtures combustion, when once initiated at any point, is almost instantaneously complete.* Moreover, he adds, they show that the specific heat of the products is very much greater at high temperatures than at low, and the extent of the difference seems to justify the view that it is the main reason of the so-called "suppression of heat." He adds that this rise in the specific heat is consistent with direct observations of that constant for $CO_2$, which have been made up to about 800° C., and which prove that it increases considerably.

**30.** In addition to these conclusions Prof. Hopkinson found certain **differences in the temperature of the gas in different parts of the vessel,** and this supports the results obtained by Prof. Burstall in his gas engine trials for the Institution of Mechanical Engineers. In experimenting with a rich mixture (air/gas = 9) Professor Hopkinson found that at the moment of maximum pressure the distribution of temperature in his vessel was roughly as follows—

| | |
|---|---|
| Mean temperature (inferred from pressure) | 1,600° C. |
| (*a*) Centre near spark . . . . | 1,900° C. |
| (*b*) 10 cm. within the wall (C, Fig. 11) . | 1,700° C. |
| (*c*) 1 cm. from wall at end (D, Fig. 11) | 1,100 to 1,300° C. |
| (*d*) 1 cm. from wall at side . . . | 850° C. |

It is explained that "at points *a*, *b* and *c* the gases can have lost but little heat at this time, and the differences of

temperature are almost wholly due to the different treatment of the gas at different places. At (*a*) it has been burnt nearly at atmospheric pressure, and compressed after burning to about $6\frac{1}{2}$ atmospheres absolute, while at (*c*) it has been first compressed to about six atmospheres as in a gas engine, and then ignited without any subsequent compression. At the point (*d*) much heat has been lost, since this is the first point on the wall reached by the flame ; the gas here is ignited when the pressure is about two atmospheres, its temperature rises instantly to 1,300° C. and at once begins to fall."

In experiments on a weak mixture of twelve volumes of air to one of gas the explosion was affected very greatly by the convection current set up, owing to the ignited gas being lighter and rising through the vessel. In the rich mixture this could not happen to the same extent as the maximum pressure was reached about a quarter of a second after firing, whilst with the weak mixture the interval was two and a half seconds and so the time for convection was much longer. The experimenter recorded that "a few centimetres below the spark the temperature will rise rapidly and then fall ; the flame reaches the wire, and is then carried upward and away from it, the wire being cooled by the current of cold, unburnt gas which follows in the wake of the ascending flame. About one second after ignition, and while the pressure is still less than 10 lb. above atmosphere, the upper part of the vessel is filled with burnt gas which is in contact with, and losing heat to, the upper half of the walls." The lower half of the gas is therefore burnt last. Finally it may be recorded that Professor Hopkinson in comparing the behaviour of rich and poor mixtures says : "It is safe to assume in dealing with a 12/1 mixture that one-fifth of a second after maximum pressure (when the loss of pressure by cooling is still less than 5 per cent.) there is present in the cylinder a mass of $CO_2$, $H_2O$, and inert gas in complete chemical equilibrium. In the 9/1 mixture this state is, of course, attained very much sooner. The difference in the behaviour of the weak and strong mixtures is wholly due to the very slow propagation of flame in the former ; in a

9/1 mixture the flame seems to travel about ten times as fast as in the 12/1 mixture."

The writer naturally has satisfaction in finding that the results of the latest and most complete experiments so far made in this subject should go so far to confirm the conclusions to which he came by a theoretic process of reasoning many years ago. In the following chapter an investigation will be made as to the modifications in the customary gas engine formulae which the adoption of variable specific heat values will require.

## EXAMPLES.

1. Answer only *one* of the following, either *A* or *B* :—

*A*. Change into horse-power the rates of conversion of chemical energy by combustion of the following :—1 lb. of kerosene per hour ; 1 cubic foot of coal-gas per hour ; 1 cubic foot of Dowson gas per hour ; 1 lb. of coal per hour. The calorific powers are, in Fahrenheit pound heat units 1 lb. of kerosene, 22,000 ; 1 lb. coal, 15,000 ; 1 cubic foot of coal-gas, 700 ; 1 cubic foot of Dowson gas, 160.

*B*. Using the calorific powers given above, calculate the efficiencies of—

(*a*) A large good condensing engine, using 2 lb. of coal per brake horse-power-hour. *Ans.* 8·47 per cent.

(*b*) A gas engine using 26 cubic feet of coal-gas per brake horse-power-hour. *Ans.* 14 per cent.

(*c*) The Diesel oil engine which is said to use 0·56 lb. of kerosene per brake horse-power-hour. *Ans.* 20·6 per cent. (B. of E., 1899.)

2. State the following amounts of energy in foot-pounds—

(*a*) A weight of 35 tons may fall vertically 15 feet.

(*b*) The Kinetic Energy of a projectile of 60 lb. moving 2,000 feet per second.

(*c*) The Calorific Energy of 1 lb. of coal, 8,500 Centigrade pound heat units.

(*d*) 30 lb. of water raised from 40° F. to 103° F.

(*e*) One horse-power-hour.

(*f*) One kilowatt-hour. (B. of E., 1906.)

3 The temperatures on two sides of an iron plate 0·5 in. thick differ by 10 Centigrade degrees, how much heat (in Centigrade pound water units) passes per square foot per second? The conductivity of iron is 0·18 in C.G.S. units (Centigrade gramme water units). (B. of E., 1906.)

4. A pound of oil contains 0·85 lb. of carbon and 0·15 lb. of hydrogen. What weight of oxygen is sufficient to produce $CO_2$ and $H_2O$ by combustion? Take the atomic weights of C, 12; of O, 16; of H, 1. If 1 lb. of oxygen is contained in 4·35 lb. of air, how many pounds of air are needed for complete combustion? *Ans.* 3·47 lb. and 15·1 lb.

5. Describe any form of coal calorimeter and its method of use.

A sample of coal is tested in such a calorimeter, and the following data are observed. Determine its calorific value.

| | |
|---|---|
| Weight of coal . . . . . | 0·015 lb. |
| Weight of water . . . . | 12 lb. |
| Water equivalent of apparatus . . | 3 lb. |
| Initial temperature . . . . | 50·1° F. |
| Final temperature . . . . | 62·5° F. |
| Time of rise . . . . . | 10 min. |
| Time of temperature to fall from 57·3° to 55·3° . . . . . . | 30 min. |

*Ans.* 13,100 B.T.U. per lb.

(Mech. Sc. Tripos, Part II, 1906.)

6. A gas engine exhausts into a calorimeter in which the gases are cooled by having water sprayed into them. The temperature of the gases after passing the calorimeter is 150° F. Assuming them to be then saturated with moisture, find the change of volume after they have been further cooled to 60° F. How would you find the heat evolved by the gases in so cooling?

(Mech. Sc. Tripos, Part I, 1905.)

## CHAPTER IV

# Thermodynamics

First Law of Thermodynamics—Second Law—Rates of Cooling of Gases—Form of Adiabatic with Variable Specific Heats—Analysis of Certain Experiments—Measurement of Cylinder Temperature—Revision of "Air Standard"—Later Measurements of Specific Heat—Flow of Heat through Metal Walls of Cylinder—Appendix.

**31. The applications of thermodynamics** to the study of gas engine problems are numerous and varied. The earlier chapter on the efficiency of cycles of operation will have afforded illustration of this, but it is proposed now to devote further attention to the matter.

Specific heat has already been defined, and indeed most students, from their work on other subjects, will be familiar in advance with calculations into which it enters. The relation between the $p$, $V$ and $T$ of a perfect gas has been given as $\frac{pV}{T}=R$, and in thermodynamic calculations it is generally necessary to assume all gases to follow this law, which it happens fortunately they very nearly do. A definition has been given in an earlier chapter of entropy or $\phi$. A new property has now to be introduced. It is called the **Intrinsic Energy**, and is known commonly as $E$.

**The Intrinsic Energy** *is the total energy actually in the substance at any moment.* Thus if heat $H$ is given to a body and mechanical work $W$ is done by the body, then the gain in intrinsic energy is $H—W$, and this may be positive or negative. So that $E=H—W$, or if small charges are dealt with, as is the most cautious procedure, it becomes $\delta E=\delta H—\delta W$.

**32.** The point has now come at which the reader may be introduced to the **two laws of thermodynamics**—

1*st Law.*—The $E$ in the gas is always the same when the gas returns to the same state; $E$ can be calculated at once if two of the variables $p$, $V$ or $T$ are known.

*2nd Law.*—The $\phi$ of the gas is always the same when the gas returns to the same state ; $\phi$ can therefore be calculated if the state be known.

From these two laws a superstructure can be raised of deductions and theorems which are useful in the solution of gas engine problems.

**33. In a perfect gas,** $E$ depends upon the temperature only, and it is equal to the product of the temperature and the specific heat at constant volume. Whether the temperature be reckoned from 0° C. or from the absolute zero is usually of little importance, as it is only the change in $E$ that has to be considered, and not its absolute amount ; so that

$$\delta E = C_v.\delta T$$

and since $$\delta E = \delta H - \delta W$$

therefore $$\delta H = C_v.\delta T + \delta W$$

or the addition of heat, $\delta H$, to the gas is balanced by $C_v.\delta T$, the gas in intrinsic energy, plus the external work done. The latter can also be written as $p.\delta V$, thus giving

$$\delta H = C_v.\delta T + p.\delta V$$

$$\text{or} \quad \frac{\delta H}{\delta V} = C_v.\frac{\delta T}{\delta V} + p \quad .. \qquad .. \qquad (1)$$

In any changes that occur to a gas it is therefore possible to find the rate of change of $H$ for unit change of volume by adding together the two terms on the right of the equation. The ratio $\frac{\delta H}{\delta V}$, which it will be observed is of the nature of a pressure, is a very important quantity in gas engine expansion and compression curves, and it is necessary to find an expression for it, which could be more quickly dealt with than equation (1).

It is required to get into a more workable form the equation.

$$\frac{\delta H}{\delta V} = C_v.\frac{\delta T}{\delta V} + p$$

Since $$\frac{pV}{T} = R$$

$$T = \frac{pV}{R}$$

and $$\frac{dT}{dV}=\frac{1}{R}\left\{p+V\frac{dp}{dV}\right\}$$

substitute and $$\frac{dH}{dV}=\frac{C_v}{R}\left\{p+V\frac{dp}{dV}\right\}+p$$

but $R=C_p—C_v$ and the ratio of $C_p$ to $C_v$ is $\gamma, \left(\frac{C_p}{C_v}=\gamma\right)$

Therefore $$\frac{dH}{dV}=\frac{C_v}{C_p—C_v}\left\{p+V\frac{dp}{dV}\right\}+p$$

or $$\frac{dH}{dV}=\frac{1}{\gamma—1}\left\{p+V\frac{dp}{dV}\right\}+p$$

or $$\frac{dH}{dV}=\frac{1}{\gamma—1}\left\{V\frac{dp}{dV}+\gamma p\right\} \quad .. \quad .. \quad (2)$$

This is found to be an easy expression to work with, and it gives $\frac{dH}{dV}$ in terms of pressure units.

**34.** The following table, part of which was calculated from a gas engine indicator card by the writer for Professor Perry's book on the *Steam Engine*, affords an illustration of the use of the formula (2)—$\gamma$ in this case was taken at the value 1·385.

| | $V$. | $p$. | $\frac{\delta p}{\delta V}$ | Average $V$. | Average $p$. | $\frac{dH}{dV}$. |
|---|---|---|---|---|---|---|
| Compression | 25 | 14·7 | | | | |
| | | | —0·96 | 22·5 | 17·1 | 5·92 |
| | 20 | 19·5 | | | | |
| | | | —1·70 | 17 | 24·6 | 13·8 |
| | 14 | 29·7 | | | | |
| | | | —3·88 | 12 | 37·5 | 14·6 |
| | 10 | 45·2 | | | | |
| Explosion and Expansion | 10 | 45·2 | | | | |
| | | | 173 | 10·1 | 62·4 | 4,760 |
| | 10·2 | 79·7 | | | | |
| | | | 218 | 10·3 | 101·5 | 6,210 |
| | 10·4 | 123·2 | | | | |
| | | | 173 | 10·5 | 140·4 | 5,230 |
| | 10·6 | 157·7 | | | | |
| | | | 120 | 10·7 | 169·7 | 3,930 |
| | 10·8 | 181·7 | | | | |
| | | | 33 | 10·9 | 184·9 | 1,590 |
| | 11·0 | 188·2 | | | | |
| | | | —22 | 11·5 | 177·2 | —20·8 |
| | 12·0 | 166·2 | | | | |
| | | | —20 | 12·5 | 156·2 | —85·8 |
| | 13 | 146·2 | | | | |
| | | | —14·8 | 14 | 131·5 | —64·9 |
| | 15 | 116·7 | | | | |
| | | | —10·5 | 16 | 106·2 | —54·5 |
| | 17 | 95·7 | | | | |
| | | | —7·5 | 18 | 88·2 | —33·8 |
| | 19 | 80·7 | | | | |
| | | | —6·0 | 20 | 74·7 | —41·5 |
| | 21 | 68·7 | | | | |
| | | | —5·0 | 22 | 63·7 | —57·1 |
| | 23 | 58·7 | | | | |

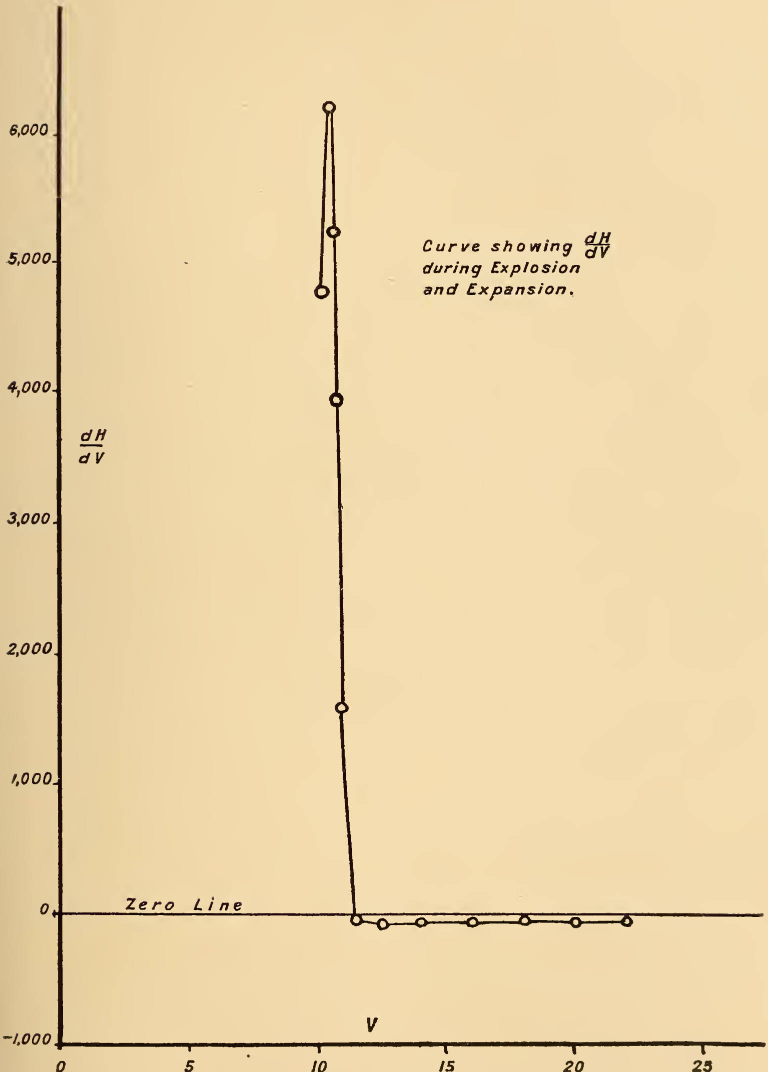



FIG. 13.—Curve of $\frac{dH}{dV}$ and $V$. It shows how the working stuff receives heat during explosive combustion and how it afterwards loses heat to the walls. Note the position of the zero line; values below it are of course negative.

These figures are plotted to scale in Fig. 13. It will be noted from the table that during compression $\frac{dH}{dV}$ is positive in every case, showing that $dH$ and $dV$ must be of the same sign. As $V$ is decreasing during compression $dV$ must be negative and therefore $dH$ also. So that during compression the gas is losing heat to the colder walls of the cylinder. The ratio of loss is not great, however, and such as it is it represents the differential effect of the cooling of the walls and the heating by contact with and radiation from the hot piston. During explosion the gas is seen, both from the table and the curve, to gain heat rapidly until the point of greatest pressure and temperature is reached, and then the curve falls rapidly and the gas begins to show a loss of heat to the cooling walls. This loss has of course been going on during the explosion also, but the effect is masked by the far greater quantities of heat then being liberated. If desired $\frac{dH}{dV}$ can be plotted on a time base.

**35.** It is often found that during compression or expansion the gas will follow the law

$$p.V^n = \text{constant} = c, \text{ say}$$

so that

$$\frac{dp}{dV} = \frac{d\,(cV^{-n})}{dV} = -ncV^{-n-1}$$

$$= -npV^n \times \frac{1}{V^{n+1}} = -\frac{np}{V}$$

or

$$V.\frac{dp}{dV} = -np.$$

Substitute this in equation (2)

and

$$\frac{dH}{dV} = \frac{1}{\gamma-1}\left\{-np+\gamma p\right\} = \frac{\gamma-n}{\gamma-1}p \quad .. \quad (3)$$

A very simple expression which can often be used to obtain results speedily. If the gas lose in $H$ during compression evidently $(\gamma-n)$ must be positive or $\gamma$ must be greater than $n$. During expansion, if the gas is losing heat, $(\gamma-n)$ must be negative or $n$ be greater than $\gamma$.

This analysis was originally due to Professors Ayrton and

Perry, and published by them in the Proceedings of the Physical Society in 1885.

It has already been explained that the loss to the cooling jacket must go on during explosion as well as expansion, and that it should be possible to draw a curve of the time rate of cooling during explosion which should have an area corresponding to the quantity of heat actually known to be carried away by the cooling water. The difficulty, however, is to know how to allow at the same time for the heating or cooling effect on the gas of the hot piston. (The writer will show presently from an analysis of certain of Professor Burstall's results what kind of effect this is.) And an additional difficulty lies in the fact that the temperature of the gas is now known to vary very considerably in accordance with its proximity to the walls.

Mr. Petavel has made some experiments on the loss of heat, $\epsilon$, per square cm. per degree Cent. by bright platinum in an atmosphere of $CO_2$, at temperatures ranging from 200° C. to 1,200° C, and at pressures ranging from 6 cm. to 228 cm. and from his results Professor Perry has published, in his book, the following rule deduced by the writer—

$$\epsilon = 1{\cdot}55 \times 10^{-8} p(1{,}000 + \theta) + 1{\cdot}67 \times 10^{-6}\theta$$

where $p$ is in pounds per square inch and the temperature is $\theta°$ C. This formula, or one of its type, has been applied to gas engine problems, but the results need not be quoted here.

**36. Effect of the variability of specific heat.** Equations (2) and (3) were obtained from premises which assumed a **constant specific heat,** and it is desirable to see how they are affected when the known variability of specific heat is allowed for. Mr. Dugald Clerk has given (see the previous chapter) the following results obtained by Messrs. Mallard and Le Chatelier in 1883—

| | | |
|---|---|---|
| For $CO_2$ | .. | $C_v = 0{\cdot}1423 + 0{\cdot}0000834\ \theta$ |
| $H_2O$ | .. | $C_v = 0{\cdot}3116 + 0{\cdot}000182\ \theta$ |
| $N_2$ | .. | $C_v = 0{\cdot}171\ + 0{\cdot}0000215\ \theta$ |
| $O_2$ | .. | $C_v = 0{\cdot}150\ + 0{\cdot}0000188\ \theta$ |

In his Report to the Gas Engine Research Committee of the Institution of Mechanical Engineers Professor Burstall

tabulates the somewhat different results published by Messrs. Mallard and Le Chatelier in 1887—

For $CO_2$ .. $C_v = 0{\cdot}1477 + 0{\cdot}000176\ \theta$
$H_2O$ .. $C_v = 0{\cdot}3211 + 0{\cdot}000219\ \theta$
$N_2$ .. $C_v = 0{\cdot}170\ \ + 0{\cdot}0000872\ \theta$
$O_2$ .. $C_v = 0{\cdot}1488 + 0{\cdot}0000763\ \theta$

From these later figures Professor Burstall has calculated for different mixtures of coal-gas and air the values of $C_v$ as shown in the curves reproduced. In 1902 * the writer

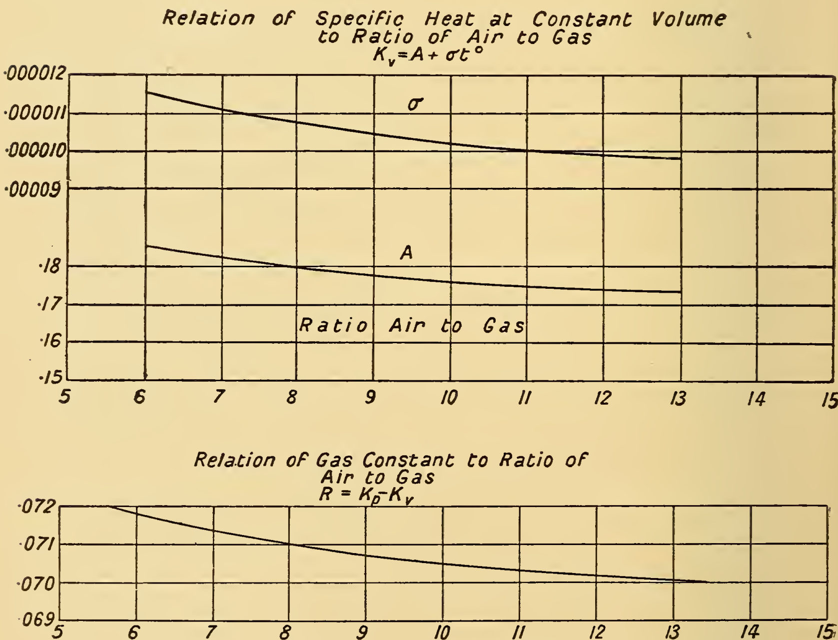



Fig. 14.—Professor Burstall's curves for specific heat constants for different ratios of air to coal-gas.

analysed certain of Professor Burstall's results, using these curves for the variable specific heats, and certain of this work is reproduced here. The discrepancies between the different measurements of specific heat are so great that the only satisfactory procedure is to keep to one set of figures,

* *Engineering*, June 27, 1902.

and use them whenever numerial results are needed. When they are not, it is best to keep carefully to symbols, which can be better interpreted when reliable results are known.

Let $C_p = \alpha + s\theta$ and $C_v = \beta + s\theta$

Now $R = \alpha - \beta$ and put $\frac{\alpha}{\beta} = \gamma_0$. i.e. the value of $\gamma$ when $\theta = 0°C$.

Then $$\delta H = C_v . \delta T + \frac{1}{J} p . dV$$

(The introduction of $J$ enables the calculation to be kept in heat units throughout.)

therefore $$\frac{dH}{dV} = C_v . \frac{dT}{dV} + \frac{p}{J}$$

Now $$\frac{pV}{T} = JR$$

so that $$T = \frac{pV}{JR} \text{ and } \frac{dT}{dV} = \frac{1}{JR}\left(p + V\frac{dp}{dV}\right)$$

and $$\frac{dH}{dV} = \frac{C_v}{JR}\left(p + V\frac{dp}{dV}\right) + \frac{p}{J}$$

This is the same equation as before, except that in this it is found convenient to keep in the constant $J$. At this point however $C_v$ must be expressed as $\beta + s\theta$.

therefore $$J\frac{dH}{dV} = \frac{\beta + s\theta}{\alpha - \beta}\left(p + V\frac{dp}{dV}\right) + p$$

$$= p\left\{1 + \frac{\beta + s\theta}{\alpha - \beta}\right\} + V\frac{dp}{dV} \cdot \frac{\beta + s\theta}{\alpha - \beta}$$

$$= \frac{1}{\alpha - \beta}\left\{\alpha p + s\theta p + \beta V\frac{dp}{dV} + s\theta V\frac{dp}{dV}\right\}$$

so that $$J\frac{dH}{dV} = \frac{1}{\gamma_0 - 1}\left\{\gamma_0 p + V\frac{dp}{dV}\right\} + \frac{s\theta}{\alpha - \beta}\left\{p + V\frac{dp}{dV}\right\} \quad (4)$$

and **this is the new expression** for $\frac{dH}{dv}$.

If $s = 0$; i.e. if specific heats were constant, equation (4) would clearly at once become equation (2).

As before, take the case where, as in compression and expansion, $pV^n = c$ very nearly.

Then as before

$$V.\frac{dp}{dV}=-np,$$

and substituting in equation (4)

$$J\frac{dH}{dv}=\frac{\gamma_0-n}{\gamma_0-1}.p+\frac{s\theta}{\alpha-\beta}\left\{p-np\right\}$$

or

$$J\frac{dH}{dV}=\left\{\frac{\gamma_0-n}{\gamma_0-1}-\frac{n-1}{\alpha-\beta}s\theta\right\}p \quad .. \quad .. \quad (5)$$

which becomes equal to equation (3) if $s=0$.

Equation (5) shows that $\frac{dH}{dV}$ is proportional to $p$ when $\theta$ is constant, and that it is a linear function of $\theta$ when $p$ is constant; provided always that all changes are regulated by the law $pV^n=$constant.

**37. Adiabatic law with variable specific heats.** If $\frac{dH}{dV}$ be zero, or, in other words, if the transformation be adiabatic, it follows from equation (4) that

$$\alpha p+s\theta p+\beta V\frac{dp}{dV}+s\theta V\frac{dp}{dV}=0$$

or

$$V\frac{dp}{dV}+\frac{\alpha+s\theta}{\beta+s\theta}p=0.$$

If $s=0$ this would become

$$V\frac{dp}{dV}+\gamma p=0$$

which on integration would give the familiar $pV^\gamma=$ constant for the adiabatic equation. Here, however, $s$ is not zero, and it is necessary to integrate

$$V\frac{dp}{dV}+\frac{\alpha+s\theta}{\beta+s\theta}p=0$$

which gives

$$p^{\beta_1}.V^{\alpha_1}.\epsilon^{s\theta}=\text{constant}$$

where $\beta_1$ and $\alpha_1$ are the constants in the equations

$$C_p=\alpha_1+sT \text{ and } C_v=\beta_1+sT.$$

This means that $\quad a_1 = a - 274\ s$
and $\quad \beta_1 = \beta - 274\ s$
so that the adiabatic equation is

$$p^{(\beta - 274\ s)} . V^{(a - 274\ s)} . \epsilon^{s\theta} = \text{constant}, \qquad (6)$$

$\epsilon$ being 2·71828, the Naperian base. This gives a higher expansion curve than $pV^{\gamma}$=constant, as is shown in the annexed diagram Fig. 15. Perhaps the best way of stating the law is

$$\beta_1 \log p + a_1 \log V + s\theta = \text{constant} \quad .. \quad .. \quad (7)$$

It will have been realized from this that if the variability of specific heats be admitted a very considerable change

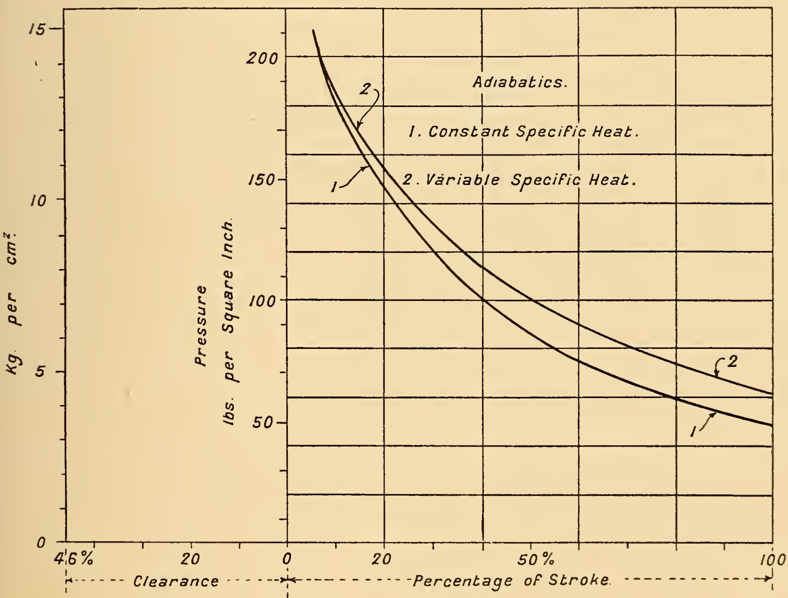



FIG. 15.—Adiabatic Curves as calculated by Professor Burstall.

must be made in the customary formula, and a good many papers and books based on the old figures must be looked upon rather as affording a mental training than as a guide to what occurs inside a gas engine cylinder.

38. In **Prof. Burstall's engine tests,** $\frac{\gamma_0 - n}{\gamma_0 - 1}$ was found to be a positive

quantity during expansion, so that had $s$ been zero, then $\frac{dH}{dV}$ would have been positive, and with an increasing $V$ this would mean that the gas was gaining heat during expansion. This could only occur when either combustion was still going on or the gas was deriving heat from the walls or piston. Naturally when faced with such a choice as this it has been usual to conclude that combustion continued until late in the stroke—a conclusion not in accord with experiments conducted in different but allied circumstances. Giving to $s$ the value already indicated has the effect of making the second term in equation (5) larger than the first, for all temperatures considered, and so leads to the more reasonable conclusion that combustion ends at or near to the point of maximum temperature, and that for the remainder of the stroke the gas loses heat to the walls at a calculable rate.

The writer has worked out some values for $\frac{dH}{dV}$ for the tests marked "D" in Prof. Burstall's investigation. Table I is composed of the leading data given in the Report, and Table II of results deduced therefrom. It is evident that the consistency of the values given in the last column of Table II—obtained as they are by the consideration of small differences—forms a very decided commentary on the extreme accuracy with which the experiments must have been conducted. It would not be easy indeed to devise a more rigorous test for experiments of this nature.

TABLE I.

1 kg. /cm.$^2$ = 0·97 atmosphere = 14·2 lb. /in.$^2$.

| D Tests | Suction Pressure kg/cm$^2$ | Suction Temperature °C. | Pressure at end of Compression kg/cm$^2$ | Temperature at end of Compression °C. | Maximum Temperature in Cycle °C. | Maximum Pressure in Cycle kg/cm$^2$ | Exhaust Temperature °C. | Exhaust Pressure kg/cm$^2$ | Jacket. Water, Temperature of Outlet °C. |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 1·00 | 143 | 8·66 | 452 | 1,437 | 13·60 | 862 | 2·73 | 65 |
| 2 | 1·00 | 140 | 8·92 | 468 | 1,509 | 18·28 | 822 | 2·65 | 62 |
| 3 | 1·00 | 132 | 8·82 | 445 | 1,442 | 14·80 | 872 | 2·83 | 62 |
| 4 | 1·00 | 128 | 8·70 | 429 | 1,454 | 14·41 | 887 | 2·90 | 64 |
| 5 | 1·00 | 115 | 8·70 | 406 | 1,372 | 14·05 | 842 | 2·88 | 66 |
| 6 | 1·00 | 110 | 8·85 | 409 | 1,245 | 13·81 | 777 | 2·75 | 60 |
| 7 | 1·00 | 98 | 8·66 | 373 | 1,145 | 12·18 | 787 | 2·86 | 66 |
| 8 | 0·95 | 84 | 8·36 | 359 | 1,094 | 11·85 | 749 | 2·72 | 66 |
| 9 | 1·00 | 84 | 8·82 | 360 | 1,023 | 12·60 | 702 | 2·73 | 63 |
| 10 | 1·00 | 69 | 8·72 | 327 | 897 | 12·00 | 637 | 2·66 | 64 |

TABLE II.

Columns 1–6 are taken from Prof. Burstall's report.

| D Tests. | $n$ in $PV^n$ for Compression | $n$ in $PV^n$ for Expansion. | During Compression. | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | 1000 $s$ | $R$ | $\gamma_0$ | $\frac{\gamma_0 - n}{\gamma_0 - 1}$ | $J\frac{dH}{dV}$ at beginning of compression | $J\frac{dH}{dV}$ at end of compression |
| 1 | 1·345 | 1·344 | 0·1834 | 0·112 | 0·0715 | 1·390 | 0·116 | 0·54 | —15·9 |
| 2 | 1·364 | 1·338 | 0·1827 | 0·111 | 0·0713 | 1·390 | 0·067 | —0·17 | —25·1 |
| 3 | 1·357 | 1·324 | 0·1811 | 0·108 | 0·0711 | 1·392 | 0·089 | 0·24 | —19·0 |
| 4 | 1·349 | 1·327 | 0·1804 | 0·107 | 0·0710 | 1·393 | 0·112 | 0 64 | —14·0 |
| 5 | 1·349 | 1·327 | 0·1785 | 0·105 | 0·0707 | 1·395 | 0·116 | 0·80 | —11·9 |
| 6 | 1·359 | 1·294 | 0·1778 | 0·103 | 0·0706 | 1·397 | 0·096 | 0·54 | —14·9 |
| 7 | 1·345 | 1·251 | 0·1765 | 0·101 | 0·0705 | 1·400 | 0·138 | 1·26 | — 5·8 |
| 8 | 1·345 | 1·245 | 0·1756 | 0·100 | 0·0702 | 1·400 | 0·138 | 1·31 | — 4·5 |
| 9 | 1·357 | 1·230 | 0·1756 | 0·100 | 0·0702 | 1·400 | 0·108 | 0·92 | — 9·2 |
| 10 | 1 350 | 1·199 | 0·1745 | 0·099 | 0·0700 | 1·401 | 0·127 | 1·32 | — 4·2 |

**39.** It is seen that $\frac{dH}{dV}$ at the beginning of compression is positive (with one exception), and as $V$ is decreasing during compression it follows that the gas is losing heat to the enclosing walls. At the end of compression, however, $\frac{dH}{dV}$ is negative without exception, showing that the gas is then gaining heat from the enclosure. Further, it is seen that this gain is greater as the maximum tempera-

Table III.

| D Tests. | Maximum Temperature in Cycle °C. | Maximum Temperature in Compression °C. | Suction Temperature °C. | End of Compression $J\frac{dH}{dV}$ | Beginning of Compression $J\frac{dH}{dV}$ |
|---|---|---|---|---|---|
| 2 | 1,509 | 468 | 140 | —25·1 | —0·17 |
| 4 | 1,454 | 429 | 128 | —14·0 | 0·64 |
| 3 | 1,442 | 445 | 132 | —19·0 | 0·24 |
| 1 | 1,437 | 452 | 143 | —15·9 | 0·54 |
| 5 | 1,372 | 406 | 115 | —11·9 | 0·80 |
| 6 | 1,245 | 409 | 110 | —14·9 | 0·54 |
| 7 | 1,145 | 373 | 98 | — 5·8 | 1·26 |
| 8 | 1,094 | 359 | 84 | — 4·5 | 1·31 |
| 9 | 1,023 | 360 | 84 | — 9·2 | 0·92 |
| 10 | 897 | 327 | 69 | — 4·2 | 1·32 |

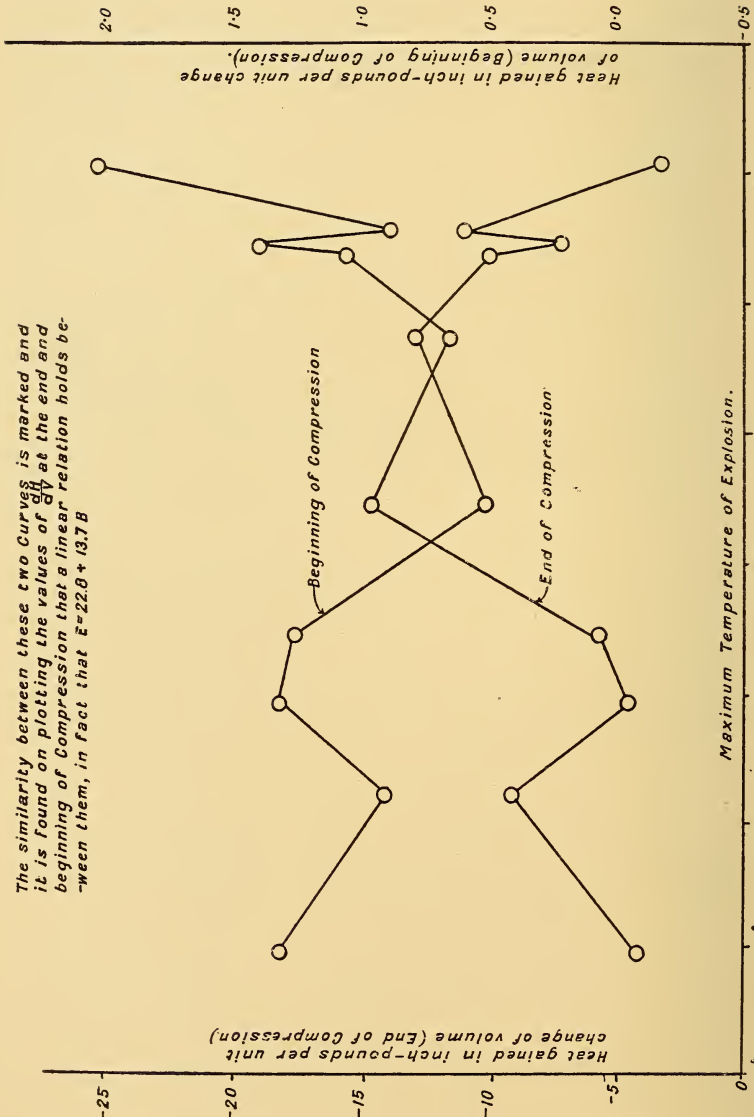

The similarity between these two Curves is marked and it is found on plotting the values of dH/dV at the end and beginning of Compression that a linear relation holds be-ween them, in fact that E=22.8+13.7B
Heat gained in inch-pounds per unit change of volume (Beginning of Compression).
Heat gained in inch-pounds per unit change of volume (End of Compression).
Beginning of Compression
End of Compression
Maximum Temperature of Explosion.
800°C
900°
1,000
1,100
1,200
1,300
1,400
1,500
1,600
−0.5
0.0
0.5
1.0
1.5
2.0
0
−5
−10
−15
−20
−25

Fig. 16.

ture of the cycle is greater. This is well seen in Fig. 16 in which the above results are shown plotted.

In fact, if, as in Table III, $\frac{dH}{dV}$ be shown tabulated in a descending scale of maximum temperatures it is at once seen that the gain of heat by the gas at the end of compression increases with increase in each of these temperatures or in fact with increase in the mean temperature of the cycle.

These results were of such an unexpected nature that it almost appeared that there must have been some slip made in the experiments themselves or in the analysis of them. The probable explanation was however considered to be, and later information has served to bear out its accuracy, that the high temperature of the face of the piston has a markedly heating effect on the gas, especially when the piston is near the beginning of the stroke, and there is a relatively thin "slab" of gas between the hot piston face on one side and the cylinder end on the other. A tabular statement will serve best to bring this point out.

(1) The gas is seen to lose heat slightly at the early part of compression.
(2) This loss is greatest when the mean temperature of the cycle is least.
(3) The gas gains heat rapidly at the latter part of the compression stroke.
(4) This gain is greatest when the mean temperature of the cycle is highest.
(5) The action of the enclosure on the gas must therefore be dual.
(6) The gas loses heat to the jacketed surface which is cooler than the gas at the moment in contact with it.
(7) The gas gains heat from the piston which during the compression stroke is always hotter than the gas.
(8) This heating is seen from Table III to occur most when the maximum temperature of the cycle is high, and as the mean temperature of the cycle rises and falls roughly in accordance with the value of the maximum temperature, the maximum effect of the heating occurs also when the mean cyclic temperature is highest, which corresponds with what would reasonably have been anticipated on this hypothesis. The observed facts and the deductions based on them do, therefore, harmonize with one another once the high temperature of the face of the piston is admitted.

**40.** A later deduction may now be added. From (6) it follows that the skin temperature of part of the jacketed walls must fall during the cycle as low as 100° C. or even less. Now Dugald Clerk has found from certain measurements that the temperature of the hottest part of the skin rises sometimes to 400° C., suggesting a temperature difference between different parts of the cylinder walls of 300° C. Further that, as regards the piston, the skin temperature must commonly exceed 500° C., and possibly exceed it a good deal, even during the compression stroke, although there has been an interval of time corresponding to more than a whole revolution since the last explosion. It may therefore be doubted whether the

temperature of the piston skin ever sinks below 500° C., and it may sometimes have an average value greatly exceeding this amount.

The deduction that the temperature of the piston is very much higher than that of the walls occurred first to the writer on making these calculations more than six years ago when for the first time theory was made to take account of variability of specific heats, and it goes far to explain these temperature variations in the mass of the gas itself which were found by Prof. Burstall by means of his electric thermometer. A simple calculation based on the thickness of the walls, the conductivity of the material and the maximum power going to the cooling water (a plate of average iron will steadily transmit per square foot of area for a gradient of 10° C. per inch about 3 h.p.) will show that the average temperature gradient cannot be large, but what its instantaneous value may be it is more difficult to say ; this point will however be treated of further shortly.

It is interesting in this connexion to work out what the actual loss in ft.-lb. per second would be according to Mr. Petavel's experimental results, remembering that it is, of course, impossible to get any exact idea of the kind of heating which occurs, owing to the large radiation from the piston masking the cooling effect of the walls and owing to the surfaces being of such different shapes and dispositions. The writer has already quoted the formula.

$$10^8 \times \epsilon = 1{\cdot}55p(1000+\theta)+167\theta \ldots \text{ for } CO_2$$

where $p$ is in lb./in.$^2$ and $\theta$ is in degrees Cent. The formula is applicable between $\theta = 200°$ C. and $\theta = 1{,}200°$ C. and up to three atmospheres pressure. Mr. Petavel has also given a formula for the emissivity, $\epsilon$, from a hot platinum wire at high temperatures and pressures as follows :—

$$\epsilon = ap^n + bp^m\delta$$

here

$\epsilon$ = emissivity in water gramme degree units per sq. cm. per sec. per deg. Cent.

$p$ = pressure in atmospheres.

$\delta$ = temperature difference between the wire and the enclosure in deg. Cent.

For $CO_2$, between $\delta = 100$ and 1,100 and between $p = 10$ and 35 this equation becomes

$$10^6 \times \epsilon = 207p^{0{\cdot}82} + 1{\cdot}50p^{0{\cdot}33}\delta \quad \ldots \ldots \ldots \ldots (8)$$

The diameter of the wire used in establishing these results was 1·106 mm., so that the area per inch of wire was 0·884 sq. cm. From this equation it is possible to get some idea of the rate of loss of heat of a mass of gas for a temperature difference of, say, 1,000°C. between the gas and the enclosure. To do so it is, of course, necessary to extrapolate, since the temperature of the gas was found by Mr. Petavel to be only 6 per cent. of that of the wire reckoning from the temperature of the enclosure as a zero. Further, this means that to the first approximation the first term in equation (8) may be neglected and that

$$10^6 \times \epsilon = 1{\cdot}50\sqrt[3]{p} \times 1{,}000 \times \frac{100}{6} \text{ very nearly.}$$

Let $p$ be 10 atmospheres, then

$$\epsilon = 53{,}500 \times 10^{-6} = 0{\cdot}0535.$$

This means that the heat loss per sec. per sq. cm. is 53·5 water-gramme-degree units. Now this is based on the consideration that as the gas was found to be 6 per cent. of the temperature of the wire, it may be taken that the loss of heat by a wire at a high temperature would be approximately the same as the loss of heat by a mass of gas, with no wire, at 6 per cent. of that temperature.

**41.** This does not appear an unreasonable hypothesis in view of the fact that it has been shown that pure radiation and conduction losses are of little importance when compared with the loss due to correction currents, currents which would certainly exist even if the wire were absent, so that in this particular example the loss per inch of "imaginary wire"

$$= 53{\cdot}5 \times 0{\cdot}884 \times 3{\cdot}09 = 146 \text{ ft. lb./sec.} = 0{\cdot}265 \text{ h.p.}$$

To use such a result as this, interesting as it may be, for gas engine work would require a further knowledge of the laws governing the radiation of hot gases to cylindrical surfaces of different diameters, and at the best it would not be strictly fair to extend it to the consideration of a non-cylindrical surface, such as a piston surface, in contact with the gas.

The above loss of 0·265 h.p. per inch length of cylinder would amount to about 2 h.p. for an average exposed length of 8 inches. The actual rate of loss during expansion is very much larger than this amount for the gas engine cylinder used by Prof. Burstall (viz. one 6 in. diam. by 12 in. stroke).

It has often been remarked that the expansion curve for a gas or steam engine almost always follows a simple law of the type

$$pV^n = \text{constant},$$

and from this it has always been shown that

$$\frac{dH}{dV} = (A + B\theta)p$$

where $A$ and $B$ are constants.

Transforming $\frac{dH}{dV}$ into $\frac{dH}{dt}$ by multiplying by $\frac{dV}{dt}$ and then expressing $\frac{dV}{dt}$ in terms of $V$ (the revolutions per minute being constant) and then eliminating $V$ by means of the equation:

$$pV^n = \text{constant}$$

it is possible to obtain an equation giving $\frac{dH}{dt}$ in term sof $p$ and $\theta$. Then since $\frac{dH}{dt}$ when divided by temperature and by area, gives the emissivity, it is possible to build up an equation showing, in terms of the emissivity, the differential heating effect due to the joint action of the cylinder walls and the piston. It is not possible, however, to follow such speculations to their end in a volume so small as this, but it is greatly to be recommended that students who are really interested in this fascinating subject should themselves pursue the matter. It will give them, perhaps—and perhaps not—valuable information, but very certainly the tackling of the many difficulties which always occur in such investigations will prove the greatest assistance in familiarizing them with the subject

and incidentally of convincing them how very little is known of what goes on in a gas engine cylinder.

**42. Experiments on Measuring Temperatures during the Cycle of Operations in a Gas Engine.**—Professor Burstall * was the first to do this. He used a platinum thermometer, and came to the conclusion † that it was impossible, owing to the fusing of the fine platinum wire before a sufficient number of observations had been taken, to make such measurements with an engine working on full load. He had therefore to experiment on an engine running light and firing but once in each twelve revolutions. The principle upon which a platimum thermometer works is that since the electrical resistance increases with the temperature in accordance with a known law, to measure the resistance of the wire is to measure its temperature at the moment. Professors Callendar and Dalby ‡ have recently made additional tests in this direction. These experimenters realized that they could not get a wire which would "stand up" to the temperature of explosion unless it was so thick that it must fail to follow the fluctuating temperatures of the gas with sufficient rapidity. They therefore decided so to arrange the apparatus that they could withdraw the fine thermometric wire from the action of the gases during explosion, and replace it for each suction and compression stroke. This was effected by fitting up the inlet valve as shown in Fig. 17. *C* is the admission valve casting, which is bolted on to the cylinder and projects inside the space provided for it. The thermometer was inserted through the spindle of the main admission valve marked *A*, which had been drilled out to receive it. In the figure the little "thermometer valve," as it may be called, is shown projecting beyond the main valve head into the cylinder. It closes with a little conical seating of its own as soon as the ignition point gets near. The thermometer leads enter through *B*, pass along the thermometer valve spindle until they arrive at the fine platinum wire which is shown at *P*. The head of the thermometer valve is connected to its spindle by the two ribs

* *Phil. Mag.*, 1895. † *Proc. I.M.E.*, 1901. ‡ *Royal Soc. Paper*, 1907.

which are made as thin as possible so that the platinum wire is not screened more than can be helped from the action of the hot gases when the thermometer valve is pushed out into action. The opening and shutting of the thermo-

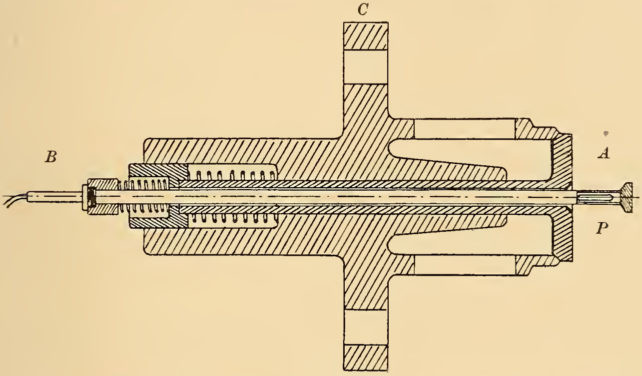



FIG. 17.—Combined Admission and Thermometer Valve (Callendar).

meter valve at the proper times is effected by suitable mechanism. The thickness of the platinum wire was $\frac{1}{1000}$ of an inch. At about 130 r.p.m. the lag of the thermometer was not more than 10° of crank angle with a temperature fluctuation of nearly 200 in half a revolution. This would correspond to a time lag of $\frac{10}{360} \times \frac{130}{60} = 0{\cdot}06$ sec., which was quite good enough for measuring anything so relatively steady as the suction temperature. As a result of such measurements it was found that the **suction temperature** varied with the conditions of running from about 95° C. at light load to about 125° C. at full load, the air temperature being about 20° C. and the jacket temperature 27° C. The following are details of two tests—

| | Test I. | Test II. |
|---|---|---|
| R.P.M. | 130 | 114 |
| Ratio air/gas | 7·1 | 5·8 |
| Atmospheric temperature | 20° C. | 21° C. |
| Jacket temperature | 27° C. | 27° C. |
| Temperature of thermometer value at 360° crank angle | 122° C. | — |
| Ditto at 26° crank angle | 111° C. | 130° C. |
| Corresponding pressure at ditto | 18·5 lb./in.$^2$ | 17·8 lb./in.$^2$ |
| Molecular contraction on combustion | 4·3 per cent. | 5·1 per cent. |

It was noted that by a curious coincidence the indicator cards from these two trials showed a practically identical expansion curve, not varying by more than 1 lb./in.$^2$ at any point. The temperatures during expansion were however far greater when using the richer mixture and the heat losses to the walls correspondingly greater, so that although much more gas is used in one case than in the other, no more h.p. is obtained, the excess heat units going to waste.

These experiments show also that the common practice of assuming the suction temperature to be 100° C. irrespective of load is an inexact one. When, as in the above experiments, the suction pressure is accurately measured, say within $+1°$ C., it is possible to calculate accurately the temperatures throughout the cycle.

**43.** It has been shown that the **adiabatic equation** with variable specific heats—*see* equation (7)—is

$$\beta_1 \log p + a_1 \log V + s\theta = \text{constant}.$$

Where $C_v = \beta_1 + sT$ and $C_p = a_1 + sT$, this could equally well be written

$$\beta_1 \log p + a_1 \log V + sT = \text{constant}.$$

Now $T = \frac{pV}{R}$ so that the equation can be written in terms of the variables $p$ and $V$ only as

$$\beta_1 \log p + a_1 \log V + \frac{s}{R} pV = \text{constant}$$

Now put $\frac{a_1}{\beta_1} = \gamma_1$; i.e. the value of $\gamma$ when $T = 0$,

$$\text{therefore } \log p + \gamma_1 \log V + \frac{s}{R\beta_1} pV = \text{constant} \quad .. \qquad (9)$$

It will be seen that this differs from the older form $pV^\gamma =$ constant (which can be written of course as $\log p + \gamma \log V = \text{constant}$) in the addition of the third term on the left of the equation. This term of course drops out when $s = 0$.

This is all based on a linear relation between specific heat and temperature, but as shown in Fig. 18 Dugald Clerk's measurements of specific heat as quoted in the preceding chapter, are nearer a parabolic relation. If however one only of the observations, that at the highest tem-

perature, on which this curve is based, be omitted a very fair straight line will lie among the rest, and this line can be sufficiently closely indicated by the equation

$$C_v = 0{\cdot}194 + 0{\cdot}051 \frac{\theta}{1{,}000} \quad .. \quad .. \quad .. \quad (10)$$

this is for a 1/9 mixture, and Burstall for such a mixture gives

$$C_v = 0{\cdot}178 + 0{\cdot}105 \frac{\theta}{1{,}000} \quad .. \quad .. \quad .. \quad (11)$$

which shows a rate of increase of about double Dugald Clerk's. Which therefore is one to choose ?

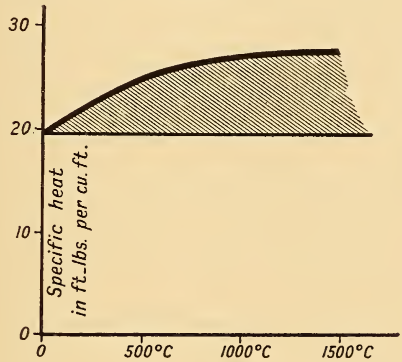



FIG. 18.—The heavy line shows Dugald Clerk's apparent specific heat curve based on the figures given on p. 55. The horizontal line corresponds to a constant specific heat and the shaded area to the difference between the two.

For $\theta = 400°$ C., they both give nearly the same result, i.e. the former gives $C_v = 0{\cdot}214$, and the latter $C_v = 0{\cdot}220$, but at 1,600° C. the former equation gives 0·275, whilst Burstall gives the much higher figure of 0·346. It is difficult to choose between these two as the systems of experiment upon which both are based are open to criticism to about the same degree. Further experiments are greatly needed.*

**44. Effect of variable specific heats on Calculation of Efficiency.** It is now important to see how the calculation of efficiency is affected by the adoption of a linear equation for the specific heat. The most important cycle is the "Constant volume" one, and in that it will be remembered (*see* par. 18) that it was found that

$$\eta = \frac{(T_2 - T_1) - (T_3 - T_0)}{(T_2 - T_1)}$$

* *See* the appendix to this chapter.

For convenience of reference Fig. 6 is reproduced in Fig. 18A.

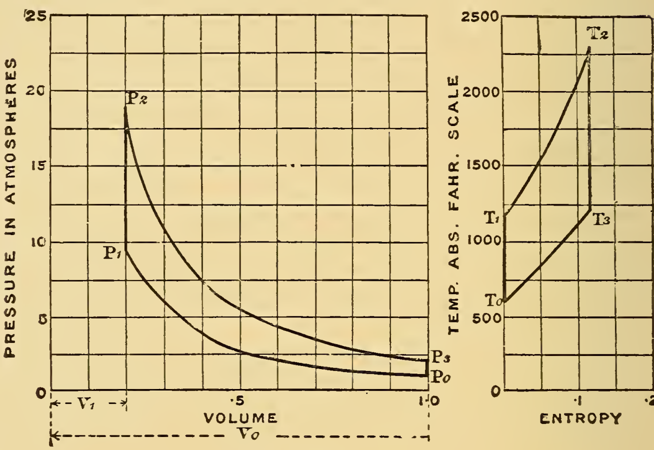



Fig. 18A.

With a variable specific heat the calculation of the thermal efficiency is far more complex. Thus heat taken in

$$=(T_2-T_1)\left(\beta_1+s\frac{T_2+T_1}{2}\right)$$

and heat rejected

$$=(T_3-T_0)\left(\beta_1+s\frac{T_0+T_3}{2}\right)$$

so that

$$\eta_0=\frac{(T_2-T_1)\left(\beta_1+s\frac{T_1+T_2}{2}\right)-(T_3-T_0)\left(\beta_1+s\frac{T_0+T_3}{2}\right)}{(T_2-T_1)\left(\beta_1+s\frac{T_1+T_2}{2}\right)}$$

where $\eta_0$ is the new efficiency.

It is now necessary to reduce this expression in some way. It still holds good that $\frac{pV}{T}=R$, and from equation (6) any adiabatic change must follow the law

$$pV^{\gamma_1}\epsilon^{\frac{S}{\beta_1}T}=\text{constant}.$$

Combining these two it follows that

$$\frac{p_2V_2}{T_2}=\frac{p_3V_3}{T_3}$$

and

$$p_2V_2{}^{\gamma_1}\epsilon^{\frac{S}{\beta_1}T_2}=p_3V_3{}^{\gamma_1}\epsilon^{\frac{S}{\beta_1}T_2}$$

so that

$$\frac{T_2}{T_3}=\left(\frac{V_3}{V_2}\right)^{\gamma_1-1}\cdot\epsilon^{\frac{S}{\beta_1}(T_3-T_2)}$$

and calling $\frac{V_3}{V_2}$, $r$ the ratio of compression as before

$$\frac{T_2}{T_3} = r^{\gamma_1 - 1} \cdot \epsilon^{\frac{s}{\beta_1}(T_3 - T_2)}$$

and
$$\frac{T_1}{T_0} = r^{\gamma_1 - 1}\, \epsilon^{\frac{s}{\beta_1}(T_0 - T_1)}$$

Substitute for $T_3$ and $T_0$ in the equation for $\eta_0$ and

$$\eta_0 = \frac{(T_2 - T_1)\left\{\beta_1 + \frac{s}{2}(T_2 + T_1)\right\} - \left[T_2 . r^{1-\gamma_1} . \epsilon^{\frac{s}{\beta_1}(T_2 - T_3)} - T_1 r^{1-\gamma_1}\, \epsilon^{\frac{s}{\beta_1}(T_1 - T_0)}\right]\left[\beta_1 + s\frac{T_3 + T_0}{2}\right]}{(T_2 - T_1)\left(\beta_1 + \frac{s}{2}(T_2 + T_1)\right)}$$

$$\eta_0 = 1 - \left(\frac{1}{r}\right)^{\gamma_1 - 1} . \left[\frac{\left\{T_2 \epsilon^{\frac{s}{\beta_1}(T_2 - T_3)} - T_1 \epsilon^{\frac{s}{\beta_1}(T_1 - T_0)}\right\}\left\{\beta_1 + s\frac{T_3 + T_0}{2}\right\}}{(T_2 - T_1)\left(\beta_1 + \frac{s}{2}(T_1 + T_2)\right)}\right]$$

Now the numerator in the square brackets must be expanded—

$$\left[\left\{T_2 . \epsilon^{\frac{s}{\beta_1}(T_2 - T_3)} - T_1 \epsilon^{\frac{s}{\beta_1}(T_1 - T_0)}\right\}\left\{\beta_1 + \frac{s}{2}\left(T_2 r^{1-\gamma_1} \cdot \epsilon^{\frac{s}{\beta_1}(T_2 - T_3)} + T_1 r^{1-\gamma_1} \cdot \epsilon^{\frac{s}{\beta_1}(T_1 - T_0)}\right)\right\}\right]$$

Now expand the exponential terms to the first two terms. (Students would do well to try the exercise of expanding to three terms.) The numerator now becomes

$$\left\{T_2\left(1 + \frac{s}{\beta_1}(T_2 - T_3)\right) - T_1\left(1 + \frac{s}{\beta_1}(T_1 - T_0)\right)\right\}\left\{\beta_1 + \frac{s}{2} r^{1-\gamma_1} \left[T_2\left(1 + \frac{s}{\beta_1}(T_2 - T_3)\right) + T_1\left(1 + \frac{s}{\beta_1}(T_1 - T_0)\right)\right]\right\}$$

$$= \left\{T_2 - T_1 + \frac{s}{\beta_1}(T_2{}^2 - T_3 T_2 - T_1{}^2 + T_1 T_0)\right\}\left\{\beta_1 + \frac{s}{2} r^{1-\gamma_1}\left[T_2 + T_1 + \frac{s}{\beta_1}(T_2{}^2 - T_2 T_3 + T_1{}^2 - T_1 T_0)\right]\right\}$$

Now the left half of this expression can be written to a sufficient degree of approximation

$$\left\{T_2 - T_1 + \frac{s}{\beta_1}(T_2{}^2 - T_2{}^2 r^{1-\gamma_1} - T_1{}^2 + T_1{}^2 r^{1-\gamma_1})\right\}$$

$$= \left\{T_2 - T_1 + \frac{s}{\beta_1}\left[T_2{}^2(1 - r^{1-\gamma_1}) - T_1{}^2(1 - r^{1-\gamma_1})\right]\right\}$$

$$= T_2 - T_1 + \frac{s}{\beta_1}(1 - r^{1-\gamma_1})(T_2{}^2 - T_1{}^2)$$

$$= (T_2 - T_1)\left\{1 + \frac{s}{\beta_1}(1 - r^{1-\gamma_1})(T_2 + T_1)\right\}$$

Therefore the whole square bracket term can be written

$$\frac{\left[(T_2-T_1)\left\{1+\frac{s}{\beta_1}(1-r^{1-\gamma_1})(T_2+T_1)\right\}\left\{1+\frac{s}{2\beta_1}r^{1-\gamma_1}\left[T_2+T_1+\frac{s}{\beta_1}(T_2{}^2-T_2T_3+T_1{}^2-T_1T_0)\right]\right\}\right]}{(T_2-T_1)\left(1+\frac{s}{\beta_1}\,\frac{T_1+T_2}{2}\right)}$$

Then neglecting terms involving $\left(\frac{s}{\beta_1}\right)^2$ this expression can be reduced as follows:—

$$\left\{1+\frac{s}{\beta_1}(1-r^{1-\gamma_1})(T_2+T_1)\right\}\left\{1+\frac{s}{2\beta_1}r^{1-\gamma_1}\left[T_2+T_1+\frac{s}{\beta_1}(T_2{}^2-T_2T_3+T_1{}^2-T_1T_0)\right]\right\}\left\{1-\frac{s}{\beta_1}\,\frac{T_2+T_1}{2}\right\}$$

$$=1+\frac{s}{\beta_1}(1-r^{1-\gamma_1})\,(T_2+T_1)+\frac{s}{2\beta_1}r^{1-\gamma_1}(T_2+T_1)-\frac{s}{\beta_1}\,\frac{T_2+T_1}{2}$$

$$=1+\frac{s}{\beta_1}\left\{(1-r^{1-\gamma_1})\;\;(T_2+T_1)+\tfrac{1}{2}r^{1-\gamma_1}(T_2+T_1)-\tfrac{1}{2}T_2-\tfrac{1}{2}T_1\right\}$$

$$=1+\frac{s}{\beta_1}\left\{T_2+T_1-T_2r^{1-\gamma_1}-T_1r^{1-\gamma_1}+\tfrac{1}{2}T_2r^{1-\gamma_1}+\tfrac{1}{2}T_1r^{1-\gamma_1}-\tfrac{1}{2}T_2-\tfrac{1}{2}T_1\right\}$$

$$=1+\frac{s}{\beta_1}\left\{\tfrac{1}{2}T_2+\tfrac{1}{2}T_1-\tfrac{1}{2}T_2r^{1-\gamma_1}-\tfrac{1}{2}T_1r^{1-\gamma_1}\right\}$$

$$=1+\frac{s}{2\beta_1}\left\{T_2(1-r^{1-\gamma_1})+T_1(1-r^{1-\gamma_1})\right\}$$

$$=1+\frac{s}{2\beta_1}(T_2+T_1)\,(1-r^{1-\gamma_1})$$

Therefore $\qquad \eta_0=1-\left(\frac{1}{r}\right)^{\gamma_1-1}\left[1+\frac{s}{2\beta_1}(T_2+T_1)(1-r^{1-\gamma_1})\right]$

And since $\qquad =1-r^{1-\gamma_1}=\eta$

it follows that $\qquad \eta_0=\eta-r^{1-\gamma_1}\frac{s}{2\beta_1}\eta(T_2+T_1)$

$$=\eta\left\{1-\frac{s}{2\beta_1}(T_2+T_1)r^{1-\gamma_1}\right\}$$

or $\qquad \eta_0=\eta\left\{1-\frac{s}{2\beta_1}(T_2+T_1)\,(1-\eta)\right\}$

which may be written

$$\eta_0=\eta\left\{1-(1-\eta)\frac{s}{\beta_1}\,\frac{T_2+T_1}{2}\right\} \quad \ldots \quad \ldots \quad \ldots \quad (12)$$

**45.** Equation (12) shows how the variability of specific heat enters into efficiency calculations. It will be noticed that $T_1$ the **compression temperature** enters into this equation, but since this temperature is so

largely affected by the compression ratio it is better to replace it by the suction temperature which is nearly constant. We can do this by reflecting that as in Equation (12), $T_1$ is multiplied by $\frac{s}{\beta_1}$, an approximation to $T_1$ may be substituted for it as follows :—

$$\frac{T_1}{T_0} = r^{\gamma_1 - 1} \cdot \epsilon^{\frac{s}{\beta_1}(T_0 - T_1)}$$

$$= r^{\gamma_1 - 1} + \text{terms in } \frac{s}{\beta_1}.$$

So that for sufficient accuracy for the present purpose

$$\frac{T_1}{T_0} = r^{\gamma_1 - 1} = \frac{1}{1-\eta}.$$

therefore
$$\eta_0 = \eta\left\{1 - (1-\eta)\frac{s}{2\beta_1}\left(T_2 + \frac{T_0}{1-\eta}\right)\right\}$$

$$= \eta\left\{1 - \frac{s}{2\beta_1}(\overline{1-\eta}.T_2 + T_0)\right\} \quad \cdots \quad \cdots \quad (13)$$

This is the important equation we are seeking.

**46.** In working out **examples** by means of this equation and getting comparative results it will suffice in most cases to give to $T_0$ some probable average value. When $T_0$ is actually known, the real value may be used, but for working out a series of results it is best to take a round figure for $T_0$. A good average figure is 400° abs., which corresponds to 127° C. The new efficiency equation then becomes

$$\eta_0 = \eta\left\{1 - \frac{s}{2\beta_1}(\overline{1-\eta}.T_2 + 400)\right\} \quad \cdots \quad \cdots \quad (14)$$

To get numerical results from this equation it is necessary to insert the specific heat constants. For the purposes of illustration it will be interesting to take the figures already given for an average working mixture as based on the work of Mr. Dugald Clerk, viz. —

$$C_v = 0{\cdot}194 + 0{\cdot}051\frac{\theta}{1{,}000}.$$

Mr. Dugald Clerk gives the weight per cubic foot at 0° C. and 760 mm. of this substance as 0·07833 lb. This enables $C_p$ to

be calculated, as the density relative to air will be $0{\cdot}07833 \div 0{\cdot}0807 = 0{\cdot}97$.

So that $$C_p - C_v = \frac{R}{J} = 0{\cdot}071$$

It follows that $$C_p = 0{\cdot}265 + 0{\cdot}051 \frac{\theta}{1{,}000}.$$

At the absolute zero of temperature $C_v$ would on this law become $0{\cdot}180$ and $C_p$, $0{\cdot}251$. So that

$$\gamma_1 = \frac{251}{180} = 1{\cdot}40.$$

If for example $r = 7$

$$\eta = 1 - \left(\frac{1}{7}\right)^{0{\cdot}40} = 0{\cdot}54$$

Substituting in our equation (14) above we have

$$\eta_0 = 0{\cdot}54\left\{1 - \frac{0{\cdot}051}{0{\cdot}360} \cdot \frac{1}{1{,}000}(0{\cdot}46T_2 + 400)\right\}$$

$$= 0{\cdot}54\left\{1 - \frac{0{\cdot}46T_2 + 400}{7{,}060}\right\}.$$

So that $$\eta_0 = 0{\cdot}54\left(0{\cdot}943 - \frac{T_2}{15{,}400}\right) \quad \ldots \quad \ldots \quad \ldots \quad (15)$$

or $$\eta_0 = 0{\cdot}51 - \frac{T_2}{28{,}400} \quad \ldots \quad \ldots \quad \ldots \quad (16)$$

**47.** Both these equations are interesting. The former shows the factor, viz.

$$\left(0{\cdot}943 - \frac{T_2}{15{,}400}\right)$$

by which the $\eta_0$ efficiency equation must be multiplied in order to get the true value. It will now be of interest to work out this result for the cases in which $T_2 = 273 + 1{,}600$ and $273 + 1{,}000$. In the former case

$$\eta_0 = 0{\cdot}51 - \frac{1{,}873}{28{,}400} = 0{\cdot}51 - 0{\cdot}07$$

$$= 0{\cdot}44, \text{ instead of } 0{\cdot}54,$$

and in the latter case

$$\eta_0 = 0{\cdot}51 - \frac{1{,}273}{28{,}400} = 0{\cdot}51 - 0{\cdot}05$$

$$= 0{\cdot}46 \text{ instead of } 0{\cdot}54.$$

It has been alleged that owing to increase of specific heat the value of $\eta_0$ *tends to a limit as the value of r increases.* An examination of equation (12)

$$\eta_0=\eta\left\{1-(1-\eta)\frac{s}{\beta_1}\frac{T_1+T_2}{2}\right\}$$

sheds a good deal of light on this suggestion, and shows it to be *untrue.* The equation can be used to do this as follows—

The total heat given up on explosion being called $H$ it follows that

$$H=(T_2-T_1)\left(\beta_1+s\frac{T_1+T_2}{2}\right) \text{ per lb. of mixture.}$$

or $$\frac{H}{\beta_1(T_2-T_1)}=1+\frac{s}{\beta_1}\cdot\frac{T_1+T_2}{2}$$

Substitute for this in the equation for $\eta_0$ and we have

$$\eta_0=\eta\left\{1+(1-\eta)\left(1-\frac{H}{\beta_1(T_2-T_1)}\right)\right\}$$

$$=\eta\left\{2-\eta-\frac{(1-\eta)H}{\beta_1(T_2-T_1)}\right\}$$

Now $T_2-T_1$ changes little even for a big change in $r$, so that it may almost be treated as constant in an expression which is multiplied by the factor $(1-\eta)$, which becomes very small as the compression is increased.

Writing this nearly constant factor

$$\frac{H}{\beta_1(T_2-T_1)} \text{ as } P$$

we have

$$\eta_0=\eta\{2-\eta-P(1-\eta)\}$$
$$=\eta\{2-P+\eta(P-1)\}$$

or $$\eta_0=(2-P)\eta+\eta^2(P-1).$$

**48.** Now, does $\eta_0$ **tend to a limit** as $\eta$ is increased up to unity? Differentiate and equate to zero,

then $$2-P+2\eta(P-1)=0$$

Therefore when $\eta=\dfrac{P-2}{2(P-1)}$ the value of $\eta_0$ is a maximum.

But

$$P=\frac{H}{\beta_1(T_2-T_1)}=1+\frac{s}{\beta_1}\cdot\frac{T_1+T_2}{2}$$

an expression but little greater than unity in almost all cases.

Therefore $\frac{P-2}{2(P-1)}$ is negative and *there is no limit to which $\eta_0$ tends as $\eta$ increases from zero upwards.* The equation

$$\eta_0=(2-P)\eta+(P-1)\eta^2 \quad \ldots \quad \ldots \quad (17)$$

happens incidentally to be one which can be used to determine values of $\eta_0$ for different values of $\eta$. The value of $P$ or $1+\frac{s}{\beta_1}\cdot\frac{T_1+T_2}{2}$ would of course require also to be known.

Putting $$\frac{s}{\beta_1}=\frac{0{\cdot}051}{0{\cdot}180}\cdot\frac{1}{1{,}000} \text{ or } \frac{1}{3{,}530}$$

we have $$P=\left(\frac{T_1+T_2}{7{,}060}+1\right).$$

Now if $(T_1+T_2)$ were, say, as high as 2,118 deg., then $P=1{\cdot}3$ and

$$\begin{aligned}\eta_0&=(2-1{\cdot}3)\eta+(1{\cdot}3-1)\eta^2\\&=0{\cdot}7\eta+0{\cdot}3\eta^2\\&=0{\cdot}3\eta(2{\cdot}3+\eta).\end{aligned}$$

If $\eta$ were 0·5, 0·7 or 0·9; $\eta_0$ would be 0·42, 0·63 and 0·86. Not too much must be built on these actual figures as they assume that $(T_1+T_2)$ is a constant and equal to 2,118 deg., which corresponds to a temperature half-way up the explosion curve of

$$\frac{2{,}118}{2}-273=1{,}059-273=786^\circ\text{ C.}$$

It is manifest that to keep this temperature fixed at 786° C. whilst the compression ratio is constantly changing represents an artificial state of affairs, but equation (17) can of course be used, when this temperature is known, in any individual case. Students are recommended to calculate out specific cases for themselves and draw a curve showing the result. For calculating out theoretic efficiencies for the constant volume cycle the author recommends the use of equation (14). When values of specific heats are better

known the calculations can be revised by inserting the more accurate constants.

**49.** Mr. Dugald Clerk in his 1907 paper before the Institution of Civil Engineers calculated how the efficiency would be affected by the adoption of the values for the specific heats found in his experiments. This he did by a graphical method and a process of successive approximations. The following table shows the results of his calculations.

| $\frac{1}{r}$ | Ideal Efficiencies. | | |
|---|---|---|---|
| | If Maximum Temperature of Cycle 1,600° C. | If Maximum Temperature of Cycle 1,000° C. | On Air Standard. |
| $\frac{1}{2}$ | 0·195 | 0·200 | 0·242 |
| $\frac{1}{3}$ | 0·286 | 0·293 | 0·356 |
| $\frac{1}{4}$ | 0·354 | 0·356 | 0·426 |
| $\frac{1}{5}$ | 0·384 | 0·394 | 0·475 |
| $\frac{1}{7}$ | 0·439 | 0·443 | 0·541 |

This table shows the very marked way in which the older expression for the efficiency is affected, when allowance is made for the variability of specific heats. It will be observed that the change is in the same direction, and of almost the same amount as that indicated by the preceding calculation. The maximum temperatures, viz. 1,600° C. and 1,000° C. are of course very low ones as they are the temperatures which occur at the highest point of the *ideal* cycle and not of the *real* one. The actual maximum temperature corresponding to Dugald Clerk's maximum of, say, 1,600° C. would of course be many hundred degrees less.

The procedure which the author suggests, viz. to calculate by formula (14) is certainly a much quicker and more general method than any graphical treatment could possibly be. Students are advised to draw up a complete table from this formula, and to consider why in certain cases the values found for the efficiency differ from those given by Dugald Clerk. In Fig. 19 is shown the result of certain experiments made by Professor Hopkinson and reported to the Institution of Mechanical Engineers in his 1908 paper. The uppermost

dotted curve is the old " Air Standard " which for the compression selected (viz. $r=6{\cdot}37$) comes out at 52·2 per cent. Under that is a line which was calculated by Hopkinson on the basis of a variable specific heat (using the figures of Holborn and Austin, and Langen). Below that again is the line

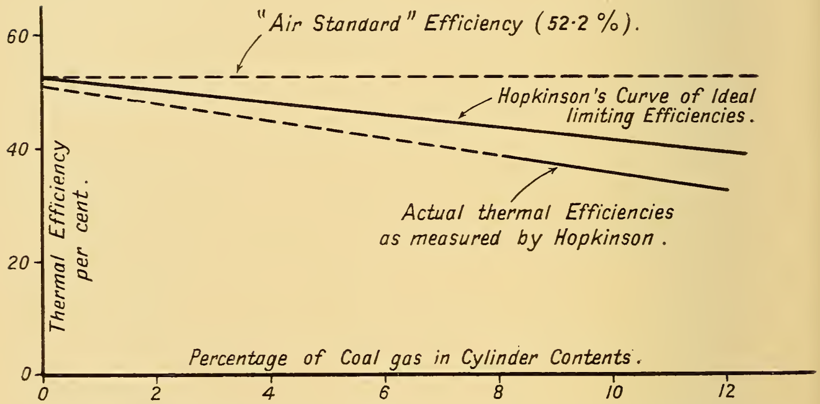



FIG. 19.—Hopkinson's measurements of actual thermal efficiency for mixtures containing from 8 to 12 per cent. of coal gas, compared with his calculated ideal limiting efficiency curve. Note the falling off in efficiency owing to increase of specific heat, as the mixture increases in calorific value.

of efficiencies as actually found. The second line was not calculated by any formula, but by an approximate, and laborious, graphical method. Taking the maximum temperatures as estimated by Hopkinson, the author's formula (14) gives an efficiency of 40·2 per cent. where Hopkinson's line gives 39·4 per cent., and 41·2 against his 42·4. The difference between the Professor's line and the author's formula is due to the calculations being founded on slightly different figures for the specific heat, and to the fact that the graphical method is only an approximation.

**50.** *Exercise.* What change would be effected in the thermal efficiency of an engine if working fluid were changed for one having a **larger specific heat** ? This is an important problem, as it not only concerns change of working fluid, but also whether it is well from the efficiency point of view to work high up the temperature scale or not.

The thermal efficiency of an engine depends on many factors, but to a first approximation it may be said to be at least proportional, for any given compression, to the efficiency as obtained from the "Air Standard" formula in which

$$\eta = 1 - \left(\frac{1}{r}\right)^{\gamma - 1}$$

where $\gamma$ = ratio of specific heats.

Now $$C_p - C_v = R \text{ and } \gamma - 1 = \frac{R}{C_v}$$

so that $$\eta = 1 - \left(\frac{1}{r}\right)^{\frac{R}{C_v}}$$

To find change of $\eta$ with respect to $C_v$, differentiate after transforming the above equation slightly

$$1 - \eta = \left(\frac{1}{r}\right)^{\frac{R}{C_v}}$$

$$\log(1-\eta) = \frac{R}{C_v}\log\frac{1}{r} = -\frac{R}{C_v}\log r.$$

therefore $$\frac{-1}{1-\eta}\,\frac{d\eta}{dC_v} = \frac{R}{C_v^2}.\log r.$$

$$\frac{d\eta}{dC_v} = -\frac{R(1-\eta)}{C_v^2}.\log r.$$

$$\frac{d\eta}{dC_v} = -\frac{R}{C_v^2}\cdot\left(\frac{1}{r}\right)^{\frac{R}{C_v}}\cdot\log r.$$

Therefore *with increase of specific heat the efficiency falls.*

This could also be written

$$\left(\frac{d\eta}{\eta}\right) = -\left(\frac{dC_v}{C_v}\right)\left\{\frac{R}{C_v}\cdot\frac{1-\eta}{\eta}.\log r\right\}$$

$$= -\left(\frac{dC_v}{C_v}\right)\left\{(\gamma-1)\frac{1-\eta}{\eta}.\log r\right\}$$

This gives the fractional change in efficiency for a given fractional change in specific heat.

If for example $\gamma = 1.40$ and $r = 10$, then for a 1 per cent. increase in $C_v$ the corresponding fractional decrease in efficiency would be

$$\frac{1}{100}\left\{(1\cdot4-1)\frac{1-\eta}{\eta}\log_e 10\right\}$$

Now when $r=10$ $\eta=0{\cdot}60$

and the above expression $=\left\{0{\cdot}40\times\frac{0{\cdot}40}{0{\cdot}60}\times 2{\cdot}30\right\}\times\frac{1}{100}$

$=0{\cdot}61$ per cent.

So that in this case the efficiency falls by **rather more than ½ per cent.** when the specific heat rises by 1 per cent.

*Exercise.* If the maximum temperature on the ideal cycle corresponding to any given real one be not known, show how formula (14) can still be employed to find the limiting efficiency for various compression ratios, provided that the **composition of the charge and its calorific value** be known.

To begin with, the heat given out on explosion =(calorific value of the gas in C.h.u. per pound) ×(weight in pounds of the gaseous mixture). In the ideal cycle this will be used in heating the gas from $T_1$ to $T_2$ so that heat absorbed = $(T_2-T_1)$ ×(mean specific heat of the gas between $T_1$ and $T_2$) × (weight of gaseous mixture). The weight of the gaseous mixture occurs in both expressions and may be cancelled out when they are equilibrated.

So that $(T_2-T_1)$ × (mean specific heat) = calorific value. Call the calorific value of the gaseous mixture per pound $K$,

then $$(T_2-T_1)\left(\beta_1+s\frac{T_1+T_2}{2}\right)=K$$

Also $T_1=\frac{T_0}{1-\eta}$ to a first approximation and in this way $T_2$ can be expressed in terms of $K$ and $T_0$ and substitution for $T_2$ made in equation (14). This will enable the value of the limiting efficiency to be calculated for any given richness of gas and ratio of compression.

**51. Later Measurements of Specific Heat.** Reference has already been made to the uncertainties of the constants in the linear equations for the specific heats of the gases used in the Internal Combustion Engine. The results which are expressed in symbols are of course true for any values of the constants, provided only that a linear law fits the facts. Measurements of specific heat made by Messrs. Mallard and Le Chatelier, have already been quoted in par. 36.

Later measurements have been made by Holborn, Austin Langen, Dugald Clerk and Hopkinson, and the author made examination of them to see whether there was sufficient agreement between these measurements which would enable the old Mallard and Le Chatelier figures to be improved upon. The result has been negative. Although it seems possible that the old experiments gave a somewhat too sharp rise of specific heat with increasing temperature, yet the discrepancies between the later measurements are too numerous to enable a decided statement to be made. For the present it suffices to use the older figures.*

It is perhaps worth while recording that Messrs. Holborn and Henning † as a result of measurement made up to 1,440° C. found that

The mean value of $C_p$ between 0° C. and $\theta$° C. was

for $N_2$ :—$0{\cdot}2350+0{\cdot}000019\,\theta$ (a straight line)

and $CO_2$ :—$0{\cdot}2010+0{\cdot}0000742\,\theta-0{\cdot}000{,}000{,}018\,\theta^2$ (a slightly curved line).

Calculated out these become—

| $\theta$° C. | $C_p$ | |
|---|---|---|
| | $N_2$ | $CO_2$ |
| 200 | ·243 | ·229 |
| 400 | ·250 | ·250 |
| 600 | ·258 | ·271 |
| 800 | ·265 | ·285 |
| 1,000 | ·273 | ·295 |
| 1,200 | ·281 | ·301 |
| 1,400 | ·288 | ·303 |

The interested student can compare these with the results previously given and with those that follow.

**52.** It is now becoming a common practice to give the **specific heat in a new form.** Instead of defining it as the quantity of heat in thermal units required to raise unit weight of gas through 1° Centigrade, it is measured as the amount of heat in ft.-lb. required to raise 1 cubic foot of the gas (measured at normal temperature and pressure—*N.T.P.*—i.e. 0° C. and 760 mm.), through 1° Centigrade. This is rather

* *See* Appendix in this chapter. † *Engineering*, January 3, 1908.

better than the old way, as it is easier to measure volumes of gases than their weights, and the "ft.-lb." form is obviously convenient.*

Take for instance nitrogen which at a given temperature has a specific heat at constant volume of 0·250. In other words, 1 lb. of nitrogen will require 0·250 C.h.u. to raise its temperature through 1° Centigrade. Now convert this into the other way of reckoning.

1 cu. ft. nitrogen at *N.T.P.* weighs 0·078 lb., so that 1 cu. ft. will require $0{\cdot}250 \times 0{\cdot}078 \times 1{,}400$ ft.-lb. to raise it through 1° Centigrade, or 27·2 ft.-lb. Mr. Dugald Clerk's recent experimental results have already been recorded in this notation, and can be compared with this figure.

As it has been found experimentally that the product of specific heat (at constant volume) by molecular weight is a constant, or nearly so, for all ordinary gases it follows that those gases will absorb about the same amount of heat per cubic foot when raised through 1° Centigrade. Thus if the value of the specific heat be given as C.h.u. per pound of gas, it is necessary to divide this figure by the number of cubic feet that go to 1 lb. of the gas in order to get the number of C.h.u. absorbed per cubic foot. This divisor will be inversely proportional to the density, and—since the density is proportional to the molecular weight—to the molecular weight also. To divide by something inversely proportional to the molecular weight is equivalent to multiplying by the molecular weight or by something proportional thereto. Therefore to get the old measurement (C.h.u. per pound) into the new one of ft.-lb. per cubic foot it is necessary to multiply by the molecular weight, and then by a constant. From this it will be seen that, to the extent that the above law is true, the value of the specific heat expressed in this way will be the same for all gases. This is obviously convenient.

**53.** It is convenient to remember that to change from the old notation to the new, all that is necessary is to

* The Gaseous Explosions Committee of the British Association suggest that the specific heat of a gas when expressed in calories per gramme-molecule should be called the "volumetric heat." "Volumetric heat" $= 3{\cdot}96 \times$ specific heat in ft.-lb. per cu. ft.

multiply by 0·078 and 1,400 in the case of $N_2$
" " 0·244 " 1,400 " " " $CO_2$
and " " 0·089 " 1,400 " " " $O_2$

Mr. Bairstow * has deduced from a study of various experimental results the table given on p. 96, which he considers

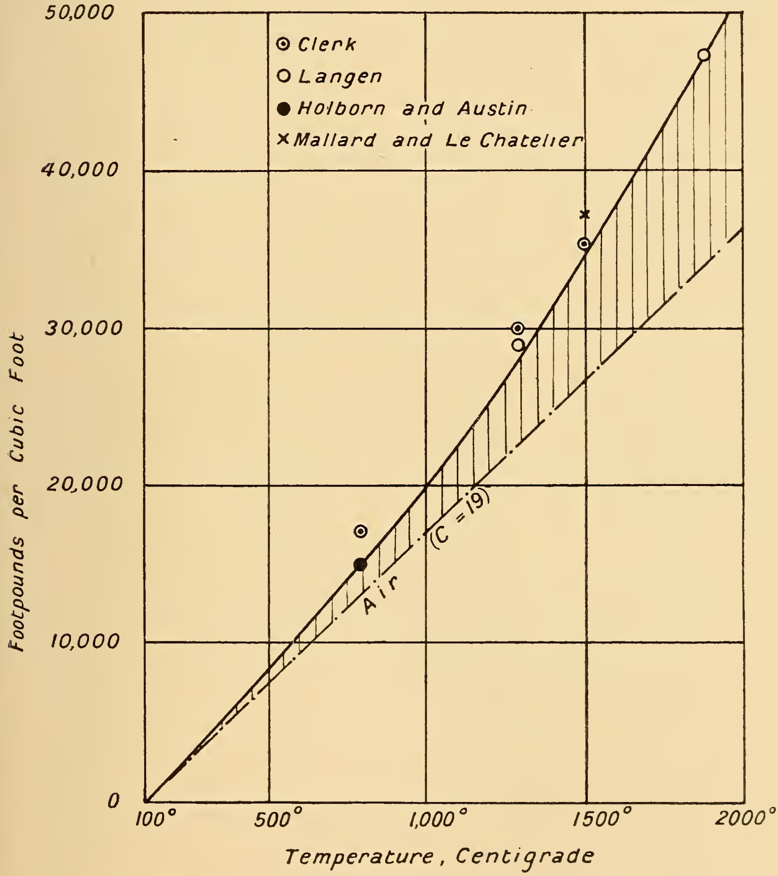



FIG. 20.—Hopkinson's Curve showing total heat (measured in mechanical units) necessary to heat gaseous mixture from 100°C. to the temperatures shown.

to show more accurately than has been done before the specific heat of a mixture produced by the explosion of nine parts of air to one of coal-gas.

* *Engineering*, June 7, 1907.

TABLE I.—THE QUANTITY OF MIXTURE CORRESPONDING TO THIS TABLE IS 1 CUBIC FOOT MEASURED AT 0 DEG. CENT. AND 30 IN. OF MERCURY.

| Temperature Centigrade. $\theta$. | Absolute Temperature. $t$. | Mean Specific Heat from 0° Cent. | Specific Heat. |
|---|---|---|---|
| | | Ft.-lb. | Ft.-lb. |
| 0 | 273 | — | 19·6 |
| 100 | 373 | 19·6 | 19·7 |
| 200 | 473 | 19·7 | 19·9 |
| 300 | 573 | 19·9 | 20·1 |
| 400 | 673 | 20·1 | 20·6 |
| 500 | 773 | 20·2 | 21·2 |
| 600 | 873 | 20·4 | 22·0 |
| 700 | 973 | 20·6 | 22·9 |
| 800 | 1,073 | 21·0 | 24·0 |
| 900 | 1,173 | 21·4 | 25·2 |
| 1,000 | 1,273 | 21·9 | 26·4 |
| 1,100 | 1,373 | 22·3 | 27·7 |
| 1,200 | 1,473 | 22·8 | 29·0 |
| 1,300 | 1,573 | 23·3 | 30·3 |
| 1,400 | 1,673 | 23·9 | 31·6 |
| 1,500 | 1,773 | 24·5 | 33·0 |
| 1,600 | 1,873 | 25·1 | 34·4 |
| 1,700 | 1,973 | 25·6 | 35·8 |
| 1·800 | 2,073 | 26·2 | 37·1 |
| 1,900 | 2,173 | 26·8 | 38·5 |
| 2,000 | 2,273 | 27·5 | 39·1 |

And based on these figures he calculates by a method of dead reckoning the following table of efficiencies.

| $r$. | Efficiency. Air Standard. | Corrected Efficiency | Corrected Efficiency Relative to Air Standard. |
|---|---|---|---|
| 4 | 0·426 | 0·335 | 0·787 |
| 5 | 0·475 | 0·384 | 0·809 |
| 6 | 0·512 | 0·417 | 0·815 |
| 7 | 0·541 | 0·445 | 0·823 |
| 8 | 0·565 | 0·470 | 0·832 |
| 9 | 0·585 | 0·490 | 0·837 |
| 10 | 0·602 | 0·508 | 0·845 |

Professor Hopkinson has endeavoured to show that after all the differences between the results of various workers

are not great. This he does in the curve shown in Fig. 20, in which he has integrated the specific heat equations so that the curve shows *the total heat in the gas* (the same mixture as that described above) at any moment. This integration of course masks the differences. At certain temperatures good agreement could be obtained between results which showed great difference in the value of $s$ but had the value of $\beta$ corrected to balance this. The shaded area in the diagram is the debatable field in which so many battles have in the past been fought.

**54.** Dr. F. Haber * has given so interesting a summary of the specific heat measurements for $CO_2$ that it is worth while quoting it here. In this summary $C$ means the product of the specific heat at constant volume, and the molecular weight (which for $CO_2$ is 44).

(*a*) $C=4{\cdot}33\left(\frac{T}{100}\right)^{0{\cdot}367}$ according to Mallard and Le Chatelier (explosion method). Basis : Explosive pressure in gas explosions at 2,000° and Regnault's numbers.

(*b*) $C=6{\cdot}5+2{\cdot}6\,\theta\times10^{-3}$ according to Langen. Basis : explosive pressures at 1,300°, 1,500° and 1,700°. The calorimetric determinations of Holborn and Austin between 0° and 800° agree with this expression.

(*c*) $C=7{\cdot}771+0{\cdot}00189\,\theta$, according to Schreber. Basis : Langen's above mentioned experiments.

(*d*) $C=6{\cdot}5+0{\cdot}00387T$ according to Mallard and Le Chatelier. Basis : experiments with the Crusher manometer.

(*e*) It may be added that Le Chatelier later concluded that the specific heats of all gases under constant pressure converged towards 6·5 at the absolute zero. The values which he considered as the most probable were—

| | |
|---|---|
| Permanent gases | $6{\cdot}5+0{\cdot}0006\,T$ |
| Water-vapour | $6{\cdot}5+0{\cdot}0029\,T$ |
| $CO_2$ | $6{\cdot}5+0{\cdot}0037\,T$. |

* *Thermodynamics of Technical Gas Reactions.* (Longmans, Green & Co.)

Dr. Haber gives a similar summary for water vapour, but reference should be made to his very valuable book for this and other important summaries of the same kind.

Holborn and Austin have also given the following interesting comparative table of values of $C_v$ at various temperatures between 0° C. and 800° C. as found experimentally by various observers.

| $\theta$° C. | Regnault. | Wiedemann. | Mallard and Le Chatelier. | Langen. | Holborn and Austin. |
|---|---|---|---|---|---|
| 0 | 0·1870 | 0·1952 | 0·1880 | 0·1980 | 0·2028 |
| 100 | 0·2145 | 0·2169 | 0·2140 | 0·2100 | 0·2161 |
| 200 | 0·2396 | 0·2387 | 0·2390 | 0·2220 | 0·2285 |
| 400 | — | — | 0·2840 | 0·2450 | 0·2502 |
| 600 | — | — | 0·3230 | 0·2690 | 0·2678 |
| 800 | — | — | 0·3550 | 0·2920 | 0·2815 |

**55.** *Exercise.*—If Van der Waal's equation for the relation between $p$, $v$ and $T$ be true, find the form taken by the equation for the specific heat at constant volume.

Van der Waal's equation was

$$\left(p-\frac{m}{V^2}\right)(V-n)=RT$$

where $m$, $n$ and $R$ are constants.

This can be written in the form

$$p=\frac{RT}{V-n}+\frac{m}{V^2}$$

which is of the type

$$p=bT+a$$

where $b$ and $a$ are functions of $V$ only.

Now $$dE=dH-p.dV$$

and $dH$ can be written, generally, as $C_v.dT+l.dV$ where $l$ is any quantity.

In the case of a gas in which the law $pV=RT$ is obeyed, $l=p$, but in the more general case now being considered we cannot assume this identity.

Therefore $$dE=C_v.dT+(l-p).dV.$$

By the 1st law of Thermodynamics this must be a complete differential, so that

$$\left(\frac{dC_v}{dV}\right)_T=\left(\frac{dl}{dT}\right)_V-\left(\frac{dp}{dT}\right)_V \quad \cdots \quad \cdots \quad (1)$$

Again $\quad d\phi=\frac{dH}{T}=\frac{C_v}{T}\cdot dT+\frac{l}{T}\cdot dV$

and by the 2nd law we must have

$$\frac{1}{T}\cdot\left(\frac{dC_v}{dV}\right)_T=\frac{1}{T}\cdot\left(\frac{dl}{dT}\right)_V-\frac{l}{T^2}$$

or

$$\left(\frac{dC_v}{dV}\right)_T=\left(\frac{dl}{dT}\right)_V-\frac{l}{T} \quad \cdots \quad \cdots \quad (2)$$

Combine equations (1) and (2),

therefore $\quad \left(\frac{dp}{dT}\right)_V=\frac{l}{T}$ or $T\cdot\left(\frac{dp}{dT}\right)_V=l.$

Differentiate and

$$T\cdot\left(\frac{d^2p}{dT^2}\right)_V=\left(\frac{dl}{dT}\right)_V-\left(\frac{dp}{dT}\right)_V$$

or by (1)

$$\left(\frac{dC_v}{dV}\right)_T=T\cdot\left(\frac{d^2p}{dT^2}\right)_V$$

Integrate and we have

$$C_v=(\text{a function of temperature only})+T\cdot\int\left(\frac{d^2p}{dT^2}\right)_T\cdot dV$$

But by Van der Waal's equation we have

$$p\doteq bT+a \text{ or } \left(\frac{d^2p}{dT^2}\right)_V=\text{zero}.$$

Therefore $C_v$=a function of the temperature only.

**56. Flow of Heat through Cylinder Walls.** A very peculiar and interesting problem is presented in the mechanism of the flow of heat through the walls of the cylinder to the cooling water. Before going into the question in any mathematical detail it is well to consider what are the dimensions of the various units concerned. The temperature inside the cylinder follows roughly a sine curve distribution in respect of time, but for simplicity assume that

the inner skin of the wall is raised suddenly to, say, 400° C. and kept at that temperature for a time equal to one stroke —say $\frac{1}{600}$ minute, i.e. 300 r.p.m.—and then lowered to a temperature of, say, 350° C, a far more violent oscillation than would be likely to occur. So for 0·10 sec. a considerable temperature difference exists between the two faces of the cylinder walls. How much is this temperature difference ? For argument's sake put it at 400—300= 100° C. Then for a 1 in. thick wall the gradient would be 100° per inch, which would give (*see* par. 40) a heat flow of about $3 \times \frac{100}{10}$ h.p. per square foot of surface, and this equals 30 h.p. during the first of the four strokes of a four-cycle engine. Now a 10 in. diameter cylinder with an 18 in. stroke will yield about 30 h.p. at normal speed and its cooling surface area would be

$$=\pi \times 10 \times \frac{18}{144} = \text{about } 4 \text{ square feet,}$$

corresponding to a heat loss on the above hypothesis of $30 \times 4$ h.p. =120 h.p., which since the engine is only a 30 h.p. one is clearly too much. What is the cause of this discrepancy ? It is that the above elementary theory assumes that a considerable heat gradient is established immediately in the metal and that no heat is absorbed in the actual heating up of the layer of metal in contact with the gas. In point of fact if the temperature were suddenly raised to 400° C. a wave of heat would pass into the iron and before it had time to travel far the engine would have made a fraction of a stroke and the skin temperature have changed its value. The manner in which this wave of heat is formed has been worked out mathematically by Fourier and is given in his *Analytical Theory of Heat.* Some very interesting problems dealing with this subject have been worked out in Professor Perry's *Steam Engine.*

**57.** A simplified treatment of the problem on mathematical lines may be given here for the benefit of those who are acquainted with elementary differential equations.

Let *OO* be the inner face of the section of the wall which can with sufficient accuracy for this problem be considered plane. Consider

what is happening at $A$, distant $x$ below the surface of the metal. Across an imaginary unit area perpendicular to the surface of the paper and to the line of flow of the heat which is in the direction of the arrow, heat will be transmitted but a part will be retained for the heating up of the substance of the lamina at $A$. At a section at distance $(x+\delta x)$ the temperature will be $(\theta+\delta\theta)$, where

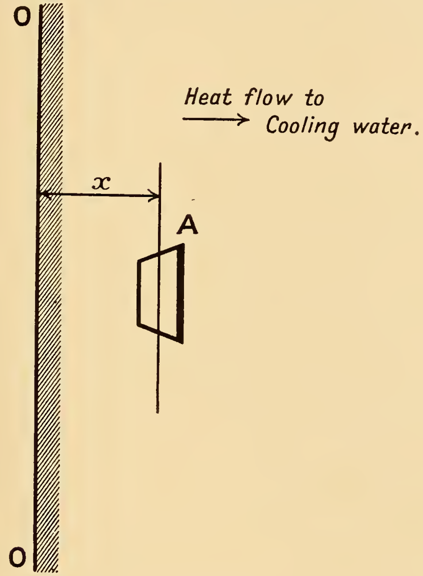



FIG. 21.

of course $\delta\theta$ is negative, at the same moment of time. Now the rate at which heat is received at the left face of the lamina contained by the two planes at $x$ and $(x+\delta x)$ is equal to $-k.\frac{d\theta}{dx}$ where $k$ is the conductivity and the additional amount which flows out per second on the other side is $\frac{d}{dx}\left(-k.\frac{d\theta}{dx}\right).\delta x= -k.\frac{d^2\theta}{dx^2}.\delta x$. Now this heat must be equal to that required to raise the temperature of the lamina between the time $t$ and the time $(t+\delta t)$, and the volume of the lamina being $(1\times 1\times \delta x)=\delta x$, it follows that the heat so absorbed must be equal to $\delta x.w.\frac{d\theta}{dt}.\delta t.s_1$, where $w$ = weight of unit volume and $s_1$ = specific heat.

Therefore

$$-k\frac{d^2\theta}{dx^2}.\delta x.\delta t = -ws_1.\frac{d\theta}{dt}.\delta x.\delta t$$

or

$$\frac{k}{ws_1}\cdot\frac{d^2\theta}{dx^2}=\frac{d\theta}{dt} \qquad \ldots \qquad (1)$$

**This is the equation for the flow of heat.** The same equation occurs in problems relating to electric conductivity, to the diffusion of liquids into each other and into many other physical applications. Its solution is therefore well known, and in this case the simplest form of it is

$$\theta = C\epsilon^{-ax} \sin(\gamma_1 t - \beta x) \quad \ldots \quad (2)$$

where $C$, $a$, $\gamma_1$ and $\beta$ are constants some of which can immediately be determined from equation (1).

From (2)

$$\frac{d\theta}{dt} = \gamma_1 C\epsilon^{-ax} \cos(\gamma_1 t - \beta x)$$

$$\frac{d\theta}{dx} = -aC\epsilon^{-ax} \sin(\gamma_1 t - \beta x) - \beta C\epsilon^{-ax} \cos(\gamma_1 t - \beta x)$$

$$\frac{d^2\theta}{dx^2} = a^2 C\epsilon^{-ax} \sin(\gamma_1 t - \beta x) + a\beta C\epsilon^{-ax} \cos(\gamma_1 t - \beta x) + a\beta C\epsilon^{-ax} \cos(\gamma_1 t - \beta x) - \beta^2 C\epsilon^{-ax} \sin(\gamma_1 t - \beta x).$$

$$= (a^2 - \beta^2) C\epsilon^{-ax} \sin(\gamma_1 t - \beta x) + 2a\beta C\epsilon^{-ax} \cos(\gamma_1 t - \beta x)$$

So that equation (1) may be written

$$\frac{k}{ws_1}\left\{(a^2 - \beta^2) C\epsilon^{-ax} \sin(\gamma_1 t - \beta x) + 2a\beta C\epsilon^{-ax} \cos(\gamma_1 t - \beta x)\right\} = \gamma C\epsilon^{-ax} \cos(\gamma_1 t - \beta x).$$

For this to be an identity

$$a^2 - \beta^2 = 0$$

and

$$2a\beta\frac{k}{ws_1} = \gamma_1$$

or

$$a = \pm\sqrt{\frac{ws_1\gamma_1}{2k}}$$

As imaginary quantities are not wanted the positive value for $a$ will be taken

So that

$$a = \beta = \sqrt{\frac{ws_1\gamma_1}{2k}}$$

Substituting in equation (2)

$$\theta = C\epsilon^{-\sqrt{\frac{ws_1\gamma_1}{2k}}\,.\,x} \,.\, \sin\left(\gamma_1 t - \sqrt{\frac{ws_1\gamma_1}{2k}}\,x\right) \quad \ldots \quad (3)$$

**58.** Now when $x = o$ the value of $\theta$ is that for the skin temperature of the metal—call this $\theta_0$

therefore

$$\theta_0 = C \sin \gamma_1 t,$$

and this suits the case in which the metal is infinitely thick and the temperature is measured from the mean temperature of the block as a zero. It shows a skin temperature which rises and falls as a simple harmonic function with an amplitude of $C$, that is to say the range of temperature in the skin is $2C$. Now imagine the wall to be in contact with a highly heated gas the temperature of which fluctuates

rapidly and unevenly. It is well known that by Fourier analysis this temperature can be represented by a series of simple harmonic functions of the time, of increasing frequency. In a gas engine the temperature of the gas rises and falls about a mean value in what is roughly a sine curve, and in any case the addition of two or three upper harmonics should make the representation very close. The effect of high harmonics at the interior part of the wall is, however, slight since it will be observed that the logarithmic decrement factor in equation (3) becomes more and more prominent as $\gamma_1$ increases in value. It will therefore be sufficiently accurate in this analysis to consider the fundamental period only and to assume that it causes in the skin of the metal a fluctuation of temperature of much the same nature but of less amplitude and with at least some lag. What this amplitude and lag will be it is almost impossible to calculate, but some idea of the former has been gained by experiment which suggests that the range in the skin of the metal is in order of magnitude always less than one-third of that in the gas. Take, however, the extreme case in which the range of temperature in this skin is actually *equal* to that in the gas. This at least will represent the *limit* of what can occur in that direction. Then $\theta = C \sin \gamma_1 t$ is the equation for the temperatures both of gas and skin.

**59. Effect at a depth.** It remains to investigate how the rest of the metal wall is affected by this great vibration in temperature in one of its faces. It is clear from equation (3) that the amplitude decreases with the depth in the metal and that a lag arises and increases at the same time. The amplitude at any point at a depth $x$ is $C\epsilon^{-\sqrt{\frac{ws_1\gamma_1}{2k}}.x}$ but $C$ is the amplitude at the surface, and therefore the fractional amplitude in the interior is $\epsilon^{-\sqrt{\frac{ws_1\gamma_1}{2k}}.x}$.

It is of interest to evaluate this expression.

We may put $w=450$ lb. per cubic foot; $s_1=0{\cdot}1$; $\gamma_1=10\pi$ for cast iron, taking the speed at 300 r.p.m.; $k=0{\cdot}01$.

So that $$\frac{ws_1\gamma_1}{2k}=\frac{450}{2}\times 0{\cdot}1\times 10\pi\times 100=70{,}500$$

or
$$\sqrt{\frac{ws_1\gamma_1}{2k}}=266$$

and the the fractional amplitude $=\epsilon^{-266x}$ here of course $x$ is in feet. If $x$ be put equal to $\frac{3}{16}$ in. or $\frac{1}{48}$ foot, the fractional amplitude $=\frac{1}{\epsilon^{5.53}}=\frac{1}{250}$ or 0·40 of one per cent., which shows that even at a depth of only $\frac{3}{16}$ inch

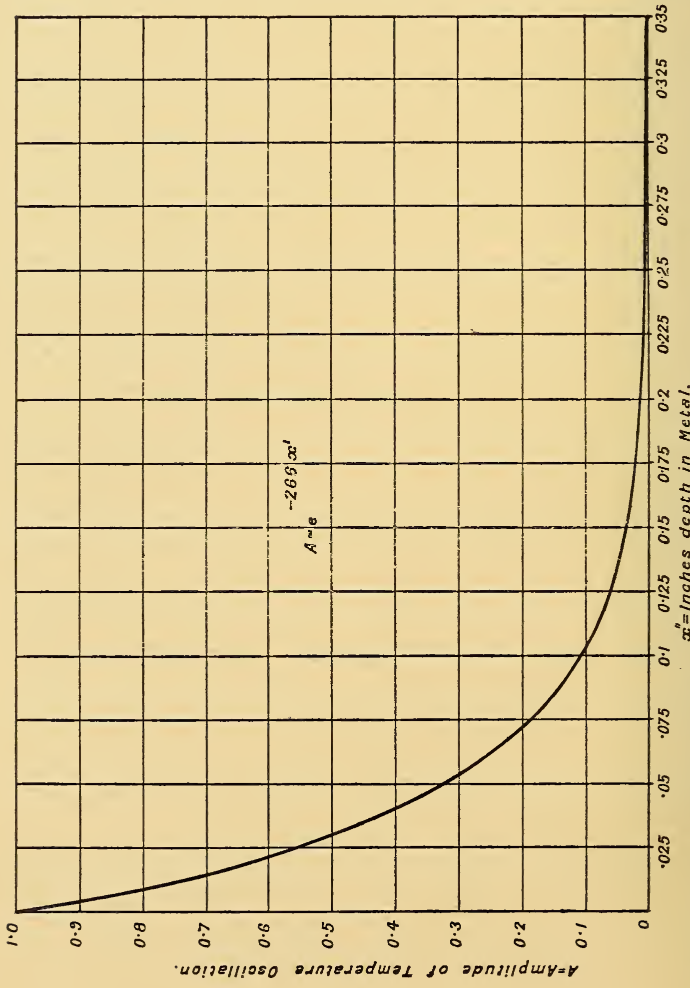



Fig. 22.—Showing amplitude of temperature oscillation in an infinite block of metal when the skin temperature oscillates above and below the mean temperature of the block.

the temperature oscillation is practically wiped out. The curve in Fig. 22 shows graphically how rapidly the oscillations decrease in amplitude. So that in assuming the wall to be infinitely thick no very far-reaching assumption was made, since for anything over $\frac{1}{4}$ in. in thickness the temperature on the water side will practically show no temperature oscillations,* and this even in so extreme a case as the above, where the inner skin is assumed to fluctuate through as great a range of temperature as does the gas itself.

**60. Practical Conclusions.** This conclusion helps to simplify matters a good deal. It shows that the *temperature gradient from face to face of the wall is practically unaffected by the oscillation in the temperature of the gas*, and that if to this sloping line, the above shallow temperature oscillations be added a representation can readily be obtained of what is actually occurring in the walls of a gas engine cylinder. The heat flow through the metal is known, as regards quantity, from the heat balance-sheet for the engine, since the heat taken away by the cooling water must be exactly equal to the flow through the walls if a steady state has been reached. The difference in temperature between outer skin and water must just be enough to enable this amount of heat to pass. What this temperature difference may be is not certainly known. But the difference between the mean temperatures of the two faces of the metal can now be calculated. Thus in a 10 in. × 18 in. engine cylinder which loses 30 h.p. continuously through the walls, of an exposed area of 4 feet, the temperature gradient will be $\frac{7\frac{1}{2}}{3} \times 10 = 25°$ C. for a wall of 1 inch thickness. If the inner skin be at an average temperature of 300° C. then the outer skin would be at 275° C., a value somewhat higher than that commonly supposed to occur.

* Since the above was written Professor Coker has published the results of some actual tests made by him, showing that at a depth in the wall of $\frac{3}{8}$ inch the range of temperature fluctuation was only $\frac{1}{500}$ part of that in the inner skin.

From the equations of par. 57 it is possible to calculate the flow of heat through the inner skin into the metal during the period of time in which the skin temperature is greater than that of the mass of the metal. Thus the heat flow at the surface must be $-k\left(\frac{d\theta}{dx}\right)_{x=0}$

$$=k\,.\,a\,.\,C\,\{\sin\gamma_1 t+\cos\gamma_1 t\}=k\,.\,a\,.\,C.\,\sqrt{2}\sin\left(\gamma_1 t+\frac{\pi}{4}\right).$$

The amount of heat flowing per sq. ft. between times $t=o$ and $t=\frac{\pi}{\gamma_1}$ must be $=\int_0^{\frac{\pi}{\gamma_1}} kaC\sqrt{2}\sin\left(\gamma_1 t+\frac{\pi}{4}\right).\,dt=\frac{2kaC}{\gamma_1}$. Inserting the values of the constants of the previous paragraph we have heat flow $=2\cdot\frac{0\cdot01\times266\times C}{10\pi}=0\cdot17C$.

Take for example a heat equivalent to $7\frac{1}{2}$ h.p. per sq. ft. of surface, then heat flow in one revolution (at 300 r.p.m.) $=7\cdot5\times\frac{550}{1400}\times0\cdot2=0\cdot59$ C.h.u., which may equate to $0\cdot17C$, giving $C=3\cdot5$ deg., or a total temperature oscillation of 7 degrees.* This indicates generally that a small oscillation in the temperature of the innermost layer of metal is quite sufficient to absorb and level the temperature oscillations which the gas tends to set up in the cylinder walls. One may conclude from this that although the temperature at any point of the walls may depend on the position of that point in the cylinder it does not sensibly vary with the time, that is to say, that at any given point the temperature in the walls remains nearly steady. The wall may therefore be considered as of two parts—the inner skin which acts as an accumulator of heat energy, rapidly abstracting it during explosion and giving it out again later; and another part, consisting of the whole of the rest of the wall, which acts as a steady transmitter of the heat fed

* Since this was written Professor Coker has published some experimental work he has done in which he found that the maximum skin temperature was not more than 4°C in excess of the mean.

into it through the inner layer. These considerations may go some way to throw light upon the problem of the cooling effect of the walls of a gas engine cylinder.

## PROBLEMS.

1. How are the pressure, volume and temperature connected in a perfect gas ?

Dry air is pumped into a closed vessel of constant volume until that pressure is 80 lb. per square inch by gauge, the temperature being 90° F. What will be the pressure in the vessel after it has remained for a considerable time in a room where the temperature is 60° F. ? *Ans.* 74·9 lb./in.$^2$ by gauge.

(Mech. Sc. Tripos, Part II, 1906.)

2. Gaseous stuff has a specific heat $K$, at constant pressure, of ·26 ; and a specific heat $k$ at constant volume of ·190. Joule's equivalent being 1393 :

(1) What is its law of adiabatic expansion ?

(2) If a pound of it is at 120° C., pressure 5,000 lb. per square foot, what is its volume ?

(3) A pound of it expands according to the law $pv^s$ constant. What is its rate of reception of heat ?

*Ans.* 7·65 cu. ft. and $(18500-1350s)$. (B. of E., 1899.)

3. Fluid expands from a point on the diagram where $p$ is represented by 1·5 inches, and $v$ by 1 inch, to a place where $v$ is 3·5 inches. According to each of the laws of expansion $p\ v$ constant, $p\ v^{1\cdot0646}$ constant, and $p\ v^{1\cdot13}$ constant, find the value of $p$ at the end of the expansion in each case. *Ans.* 0·428 ; 0·395 ; 0·376 ins.

(B. of E., 1900.)

4. The law of cooling in Dugald Clerk's gas explosion experiments is $p=\frac{31}{\sqrt{t}}$ where $t$ is time in seconds and $p$ is pressure in lb./in.$^2$. Calculate the rate of loss of heat per second, i.e. $\frac{dH}{dt}$. Given that $C_v=\beta+s\theta$.

5. In a gas engine diagram the expansion curve usually lies above the adiabatic expansion curve, showing that if

the working substance be a perfect gas it must be receiving heat during the expansion, yet in fact much heat is withdrawn from the cylinder walls by the cooling water. What do you regard as the most probable explanation of this? Give some account of the arguments and experimental evidence which lead you to prefer your explanation to others that have been adduced.

(Mech. Sc. Tripos, Part II, 1904.)

6. A gas expands so that $pv^n$ is constant. Show that if $n$ is equal to the ratio of the specific heat at constant pressure to the specific heat at constant volume the expansion is adiabatic. (Mech. Sc. Tripos, Part I, 1898.)

7. The piston of an air compressor displaces 8 cubic feet per stroke and makes 120 strokes per minute. It takes in atmospheric air at 60° F. and compresses it, according to the law $PV^{1\cdot 25}$=constant, up to 75 lb. gauge pressure; finally delivering it at this pressure into a reservoir. Assuming no slip past the valves, no loss of head through them, and in the first place no clearance, calculate the work done upon the air in foot-pounds per minute, and the temperature at which it enters the reservoir. In the second place, if the clearance were 10 per cent. of the piston displacement, how would the work done per minute and the volumetric efficiency of the compressor be affected? Atmospheric pressure=14·7 lb. per square inch.

*Ans.* 2·26 cu. ft., 19·8 B.T.U. and —0·083.

(Mech. Sc. Tripos, 1906.)

8. In a gas engine release occurs at seven-eighths of the stroke and at a pressure of 40 lb. per square inch absolute. The clearance space is a quarter of the total cylinder volume. The engine works on the Otto cycle, explodes every time and is not scavenged. The mixed gas and air just before being drawn into the cylinder on the suction stroke has a temperature of 100° C. Estimate the temperature of the charge filling the cylinder at the end of the suction stroke.

In making your estimate you will probably assume that gases before and after explosion behave as the same perfect gas. How far is this assumption correct? Illustrate the possible errors in estimates of temperature based on this

assumption by finding their amount in the case of a mixture of one volume of hydrogen and five of air. *Ans.* 3·4 and $3{\cdot}8 \times 10^6$ ft.-lb./min. (Mech. Sc. Tripos, Part II, 1904.)

9. It is found that the area of the diagrams of the Crossley engine in the laboratory averages about 20 per cent. bigger when the engine is running light than when it is fully loaded. Explain this.

When the engine is running fully loaded the temperature of the exhaust gases left in the clearance space at the end of the exhaust stroke is 700° C., and the temperature of the gas and air sucked in just before they enter the cylinder is 100° C. The clearance space is a quarter of the total cylinder volume (including clearance space). Show that the temperature of the gases filling the cylinder at the end of the suction stroke will be 170° C. Assume that no heat is lost to or gained from the cylinder walls during suction, that the pressure inside the cylinder is the same as that of the atmosphere, and that the specific heat of the exhaust gases, and of the incoming charge is the same constant quantity. (Mech. Sc. Tripos, Part I, 1904.)

10. The equation for the flow of heat in the walls of an engine cylinder may be assumed to be

$$\frac{k}{c}\,\frac{\delta^2 V}{\delta x^2} = \frac{\delta V}{\delta t},$$

$k$ and $c$ being the conductivity, and thermal capacity per unit volume, of cast iron, and $V$ the temperature at time $t$ at the distance $x$ from the surface. Show that the solution of this equation is $V = V_1 \epsilon^{-mx} \cos(\theta - mx)$ for a simple harmonic variation of surface temperature of semi-range $V_1$; $\theta$ being the angle described by the crank in time $t$, its angular velocity being $2\pi n$.

Hence find the range at any depth $x$; prove that the value of the index co-efficient, $m$, is $\sqrt{(\pi nc/k)}$; and find the heat absorbed in thermal units per square foot of wall surface per period, the semi-range at the surface being $V_1$.

(Mech. Sc. Tripos, 1906.)

11. Prove the formula for efficiency in the hypothetical Otto Cycle (the diagram being two adiabatics and two constant volume lines), showing how efficiency is greater as

clearance is less. In what way does this hypothetical diagram differ from reality? If it differs greatly, why are such calculations of any use? (B. of E., 1906.)

12. Given a gas engine diagram, show how we may draw a diagram showing the rate (1) per cubic foot change of volume, (2) per second, at which the stuff receives heat from the beginning of the compression to the release.

You may assume an infinitely long connecting rod. (B. of E., 1900.)

13. An air compressor pumps air in a steady stream into the lower part of a reservoir against a pressure of 500 lb. per square inch absolute. The reservoir is partly filled with water and is maintained, by an outside source, at a constant temperature of 300° F. An equal stream of air passes from the upper part of the reservoir through a reducing valve, carrying with it water vapour and suspended water and goes to supply a system of air motors. Find what portions of the pressure in the reservoir are due to air and water vapour respectively, and how many pounds of air are mixed with each pound of vapour in the upper part of the reservoir.

If the pressure beyond the reducing valve is $p$ lb. per square inch absolute, write down the equations which determine the state of the mixture after passing the valve, assuming that each pound of vapour carries 0·3 lb. of suspended water, and that the vapour does not become superheated.

14. A mass of unequally heated perfect gas is enclosed in a vessel whose walls are impervious to heat. Prove that the pressure of the gas remains unchanged during the equalization of the temperature by connexion and conduction. (Mech. Sc. Tripos, Part I, 1904.)

15. Air flows through an orifice from a reservoir in which the pressure is $p$ lb. per square foot and temperature $t$ into a region of lower pressure—heat being neither received nor rejected during the operation. Obtain an expression for the maximum discharge in pounds per second, in terms of $p$, $t$, the effective area of the orifice, and the ratio of the specific heats.

Are the general conclusions arrived at by theory verified in practice? If not, state what the experimental results obtained really are, and point out where the theory probably fails. (Mech. Sc. Tripos, Part II, 1904.)

16. Air is compressed adiabatically into a receiver of $V$ cubic feet capacity to $m$ times the atmospheric density. Show that if $p$ be equal to the atmospheric pressure, the work expended is

$$pV\left(\frac{\gamma m^{\gamma}-\gamma m}{\gamma-1}+1\right)\text{foot-lb.}$$

$\gamma$ being the ratio of the specific heats of air.

(Mech. Sc. Tripos, Part I, 1904.)

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# APPENDIX

As this book goes to press, the Gaseous Explosions Committee of the British Association publish their first report. After considering all the available specific heat measurements they present a curve of $C_v$ and $\theta$, which they consider to represent values accurate to 5 per cent. The following equation fits this curve very well.

$$C_v = 0{\cdot}172 + 0{\cdot}075\ \frac{\theta}{1000}$$

This would make equation (14) become

$$\eta_0=\eta\left\{1-\frac{1}{4000}\ (\overline{1-\eta}\,.\,T_2+400)\right\}$$

and equation (15)

$$\eta_0=0{\cdot}60\left(0{\cdot}90-\frac{T_2}{10{,}000}\right).$$

Also it is worth noting that the efficiencies of par. 47 would become 0·43 and 0·46 in place of 0·44 and 0·46.

The further report of the Committee will be awaited with much interest.

*SECTION II*

# GAS ENGINES AND GAS PRODUCERS

## CHAPTER V

# The Gas Engine

TYPES OF GAS ENGINE—METHODS OF IMPROVING THEIR EFFICIENCY—INDICATORS, OLD AND NEW—HEAT BALANCE SHEETS—ENGINE TESTS — GOVERNING — CYCLIC IRREGULARITY — BALANCING.

**61. Types of Gas Engine.** The several thermodynamic cycles upon which gas engines are capable of working have already been described, but it may be said at once that practically all gas engines now being built are designed to work on the **constant volume cycle,** or as near thereto as can be effected. Students will also hear of other cycles such as the **Otto** and the **Clerk** cycles. These names refer to the cyclic operation of the exhaust and inlet valve gear and *not* to the thermodynamic ideal to which it is desired to make combustion conform. The Otto cycle consists of **four strokes:** the **admission** stroke when the piston is moving outwards, the **compression** stroke when it returns, the **expansion** stroke which occurs after explosion has taken place, and the fourth stroke, generally known as the **scavenging** stroke, when the burnt gases are pushed out of the cylinder. As each cycle includes two revolutions of the engine the valves are operated from a cam shaft which rotates at half the speed of the main shaft and is therefore called the **half-time-shaft.** Examples of the Otto cycle are found in nearly every type of engine now built. In the Clerk cycle there are only **two strokes,** the explosion stroke and the compression stroke. This cycle will now be described in greater detail.

In addition to the actual engines illustrated in this chapter the author shows, also, illustrations of the more important engine details, for many of which he is indebted to the kindness of the National Gas Engine Co.; and for

others, to the several makers of the engine types described.

**62.** The majority of gas engines * at present in use work on the Otto cycle, but a considerable number of the larger

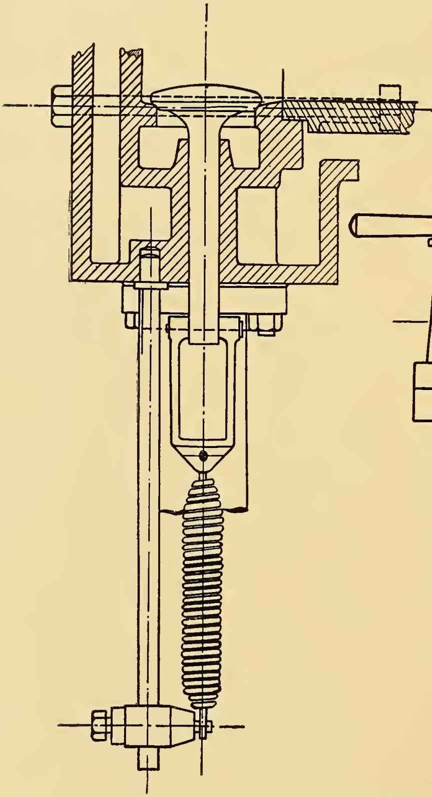


FIG. 23. — Exhaust valve of a National Gas Engine. Mushroom type. Four Cycle Engine.

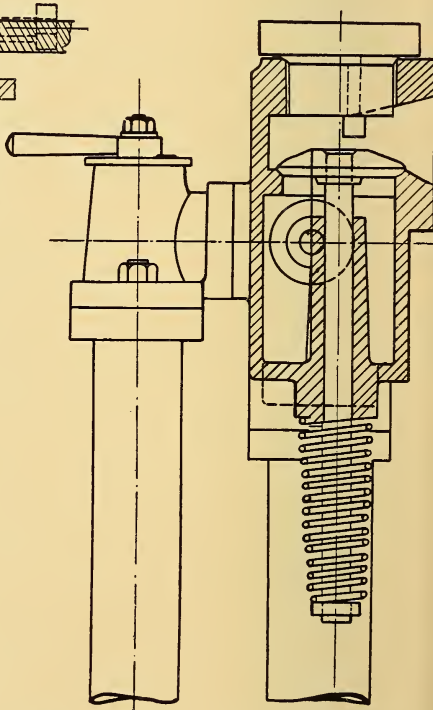


FIG. 24.—Inlet valve of a National Gas Engine. Mushroom type. Four Cycle Engine.

sizes of engine do not. Among the latter are the **Koerting** engine, made in this country by Messrs. Mather & Platt; and the **Oechelhäuser** engine, which is manufactured in England by Messrs. Beardmore. Illustrations of both these types are shown. In each case the exhaust passes through ports in the cylinder walls which are overrun

* All are water-jacketed. It is only the smallest petrol engines that rely on air cooling.

by the moving piston. The **Koerting** piston is made long and the ends of the cylinder are coned. At these coned ends are placed the inlet valves, through which the working charge is *pumped* by two pumps, one for air and one for gas. The engine is a double-acting one and every stroke is a working stroke just as in a steam engine. The great length of the piston prevents the exhaust ports being overrun until

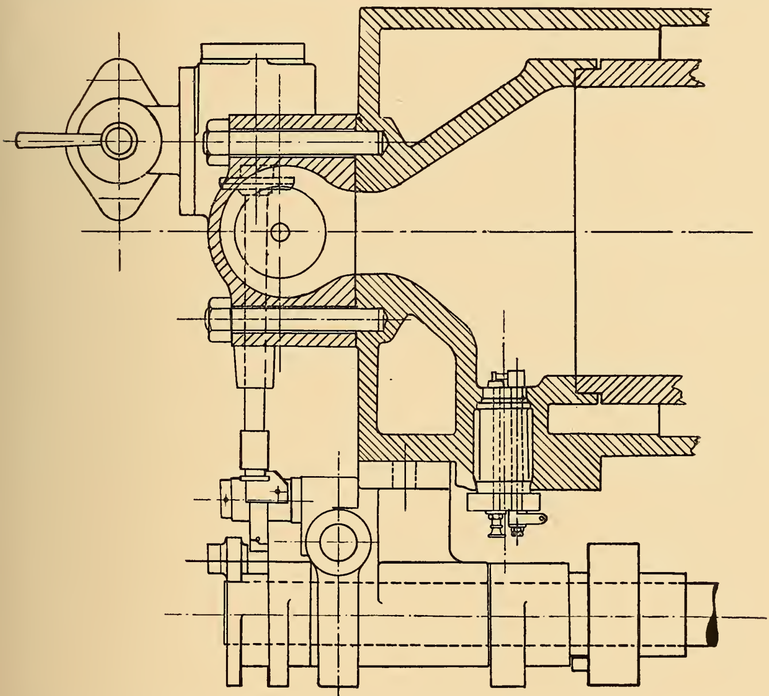


FIG. 25.—Admission end of Cylinder of National Gas Engine. Note the conical interior and fitting of cylinder liner. Note also the cams on half-time-shaft for operating valves. Four Cycle Engine.

the end of the stroke, whether the piston is moving from left to right or right to left. Once the exhaust port is uncovered the gases pass away, and are helped on their passage by the air which is then being admitted in the coned ends of the cylinder. When this has gone on for a short time, gas is admitted also and the mixture is ready for compression on the further motion of the piston. The time taken for

the crank to pass the few degrees of slow piston motion on each side of the dead centre affords opportunity for these

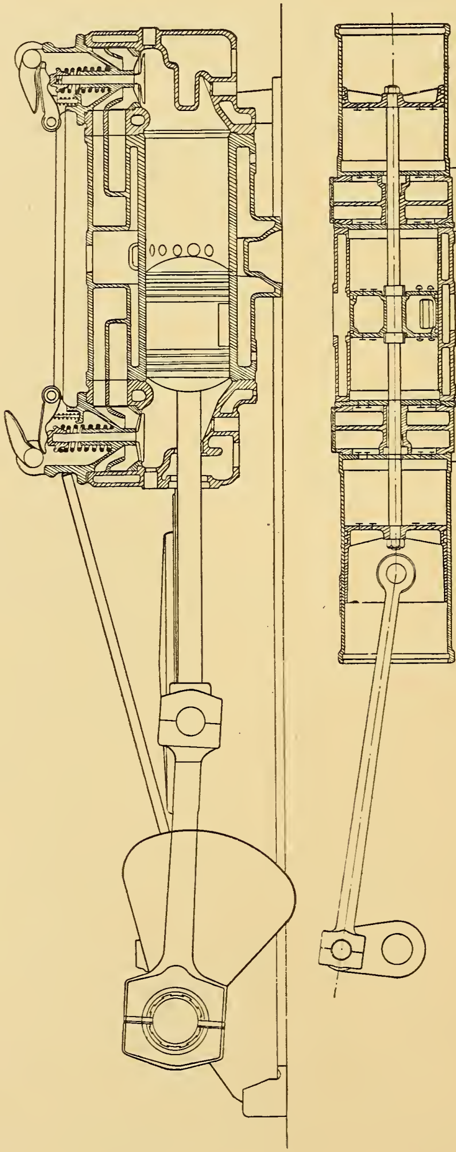


FIG. 26.—Longitudinal Section of Two Cycle Engine (Koerting)—new type, by Messrs. Mather and Platt. Power cylinder above. Gas and Air pump below. Note inlet valves at each end and length of piston in power cylinder.

operations to take place. An obvious difficulty about this method of working is that some of the incoming gas may be caught up and pass away with the exhaust products and so be lost. This reduces economy, but is of little importance when working with what are known as *waste gases**, as these are produced in immense volume and at practically no cost. Attention has been given in the previous chapter to the question of piston and cylinder wall temperatures, and it will therefore be readily understood that in such a cycle

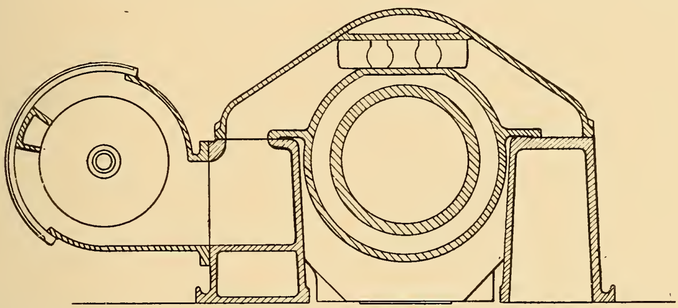


FIG. 27.—Cross Section of Two Cycle Engine (Koerting)—new type, by Messrs. Mather and Platt. Compare with Fig. 26.

as this the heating effect of explosions so closely following each other will be severely felt and high temperatures are likely to be reached by all parts open to the gases.

**63.** The **Oechelhäuser** engine resembles the Koerting inasmuch as it has ports in the cylinder walls which are overrun by the pistons, but the method of working is quite different, as in this type each cylinder has two pistons which move inwards and outwards together, so producing a well-balanced motion. The joint centre of gravity of the two pistons does not move. One piston operates directly on its crank and the other through return connecting rods on to another crank placed 180° from the former one. As will be seen from the illustration (Fig. 31), the piston on the left overruns first the exhaust ports, thus enabling the exploded gases to leave the cylinder. This operation is then helped by the right-hand piston overrunning the air inlet ports so that air rushes in and aids the exit of the waste products.

* *See* Chapter VII.

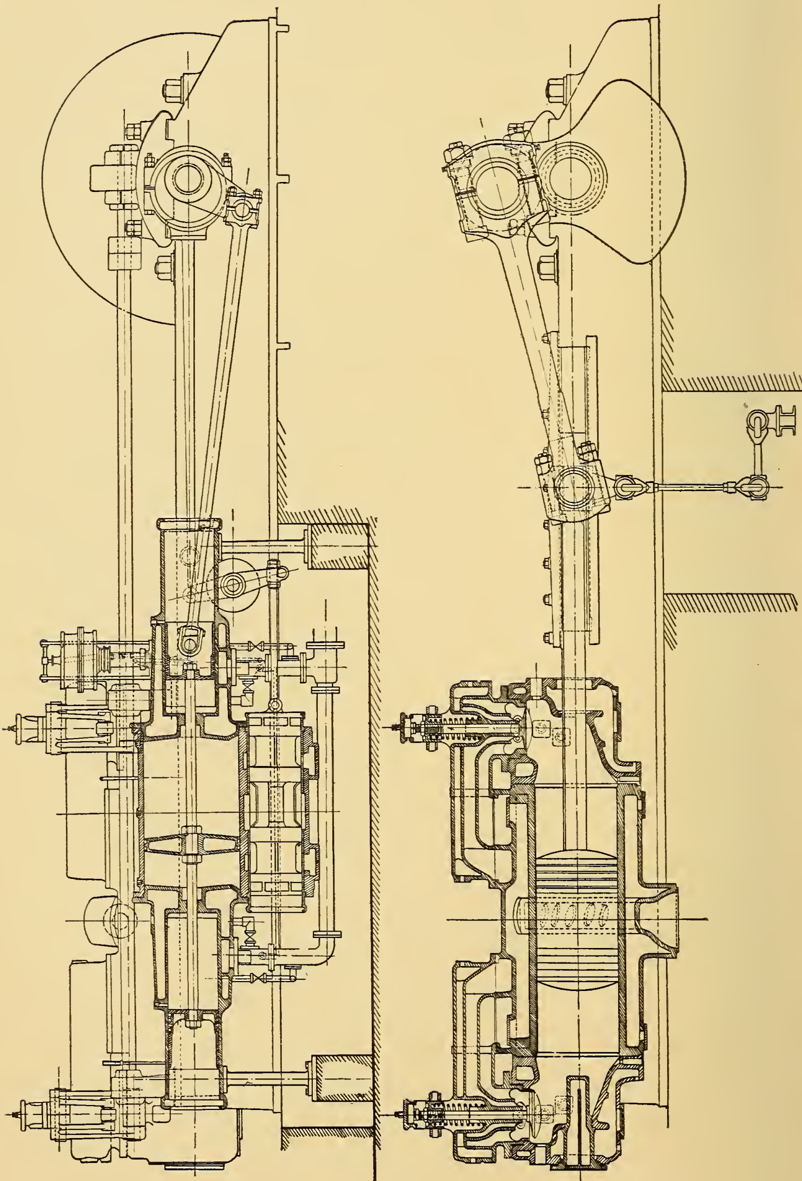


FIG. 28.—Longitudinal Section of 400 B.H.P. Koerting Engine by Mather and Platt. Note arrangement of Gas and Air Pump above.

Then the right-hand piston in its further motion overruns the gas inlet ports and so a proper mixture is admitted, which, when the two pistons move in together, gets compressed for the next explosion. This gives an explosion for every time the pistons separate, or an explosion every revolution. The engine is provided with separate pumps for the

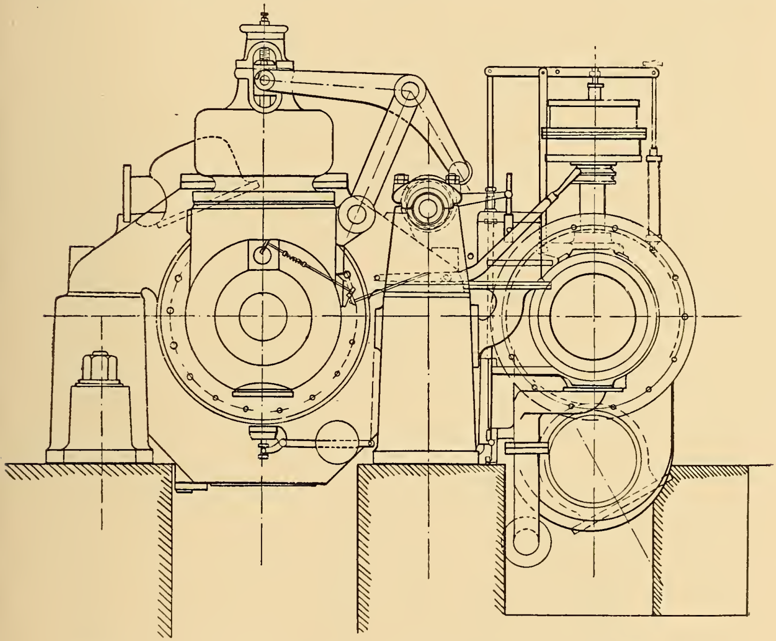


Fig. 29.—Cross Section of 400 B.H.P. Koerting Engine by Mather and Platt.

air and gas. That there are many difficulties to be overcome in the construction and working of large gas engines, particularly those working on the Clerk cycle, is shown by the unfortunate history of the generating station for lighting and tramway work which was established some years ago at Johannesburg. The plant was operated with Oechelhäuser gas engines driven by gas from Poetter producers using Transvaal coal. The electric portion of the plant consisted of Siemens generators, giving a total output of 8,100 K.W. From the very beginning many difficulties arose in the operation of this plant, mainly on account of the

lack of requisite experience in the manufacture and operation of similar installations. The cost per unit is reported

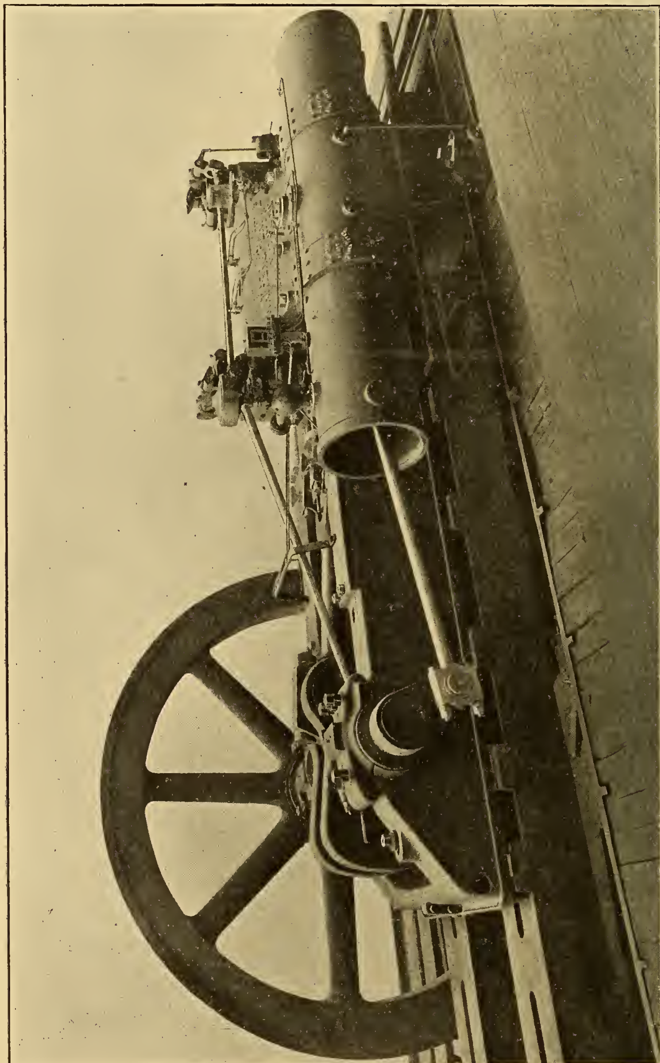


FIG. 30.—300 B.H.P. Koerting Gas Engine (Mather and Platt). Example of latest construction. Note inlet valves operated by eccentric on main shaft.

to have been higher than was originally anticipated, and it is understood that the plant has since been shut down.

This occurrence must not be taken to show that Oechelhäuser engines are unsuitable for heavy work, since in numerous places on the Continent engines of this make are working well and giving entire satisfaction to their owners. The largest installation of gas engine plant in this country is at present that at the Cargo Fleet Iron Co.'s works at Middlesbrough. It consists of six **Cockerill** engines, built by Messrs. Richardsons, Westgarth & Co., of 900 h.p. each, or a total of 5,400 h.p. These engines work on the **Otto cycle,** but by having double-acting tandem cylinders the crank gets just as many impulses as in a double-acting steam engine.

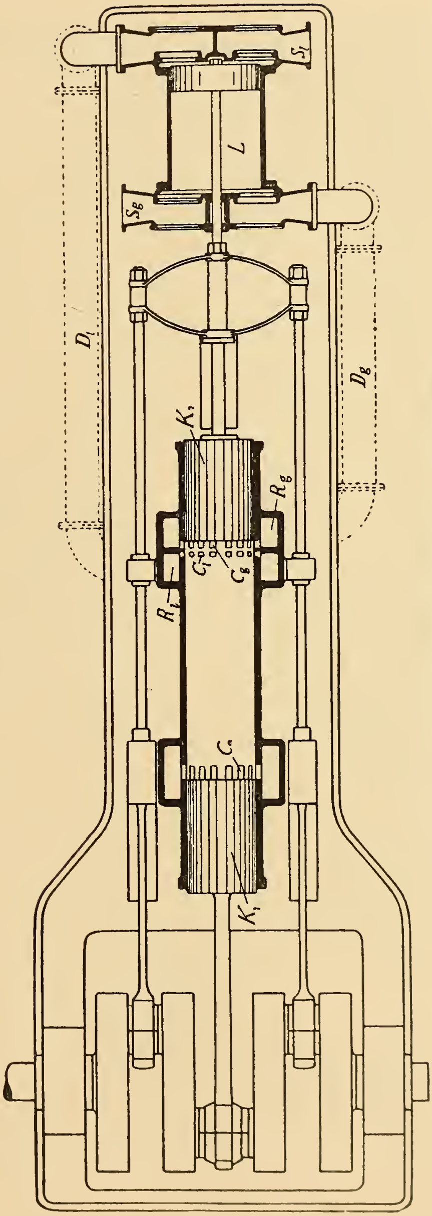


Fig. 31.—Arrangement of Oechelhäuser Gas Engine.

**64.** Having now gained some idea of the manner of working of a few of

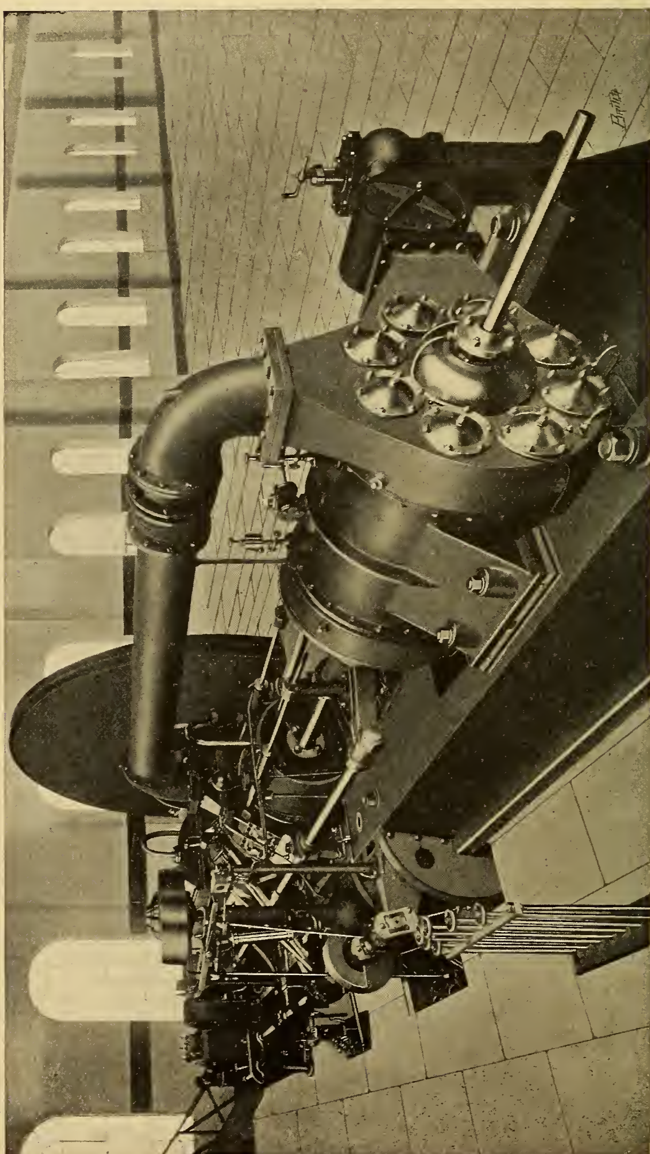


FIG. 32.—500 H.P. Oechelhäuser Engine coupled to Dynamo, giving 440 volts and 850 ampères at 105 r.p.m.

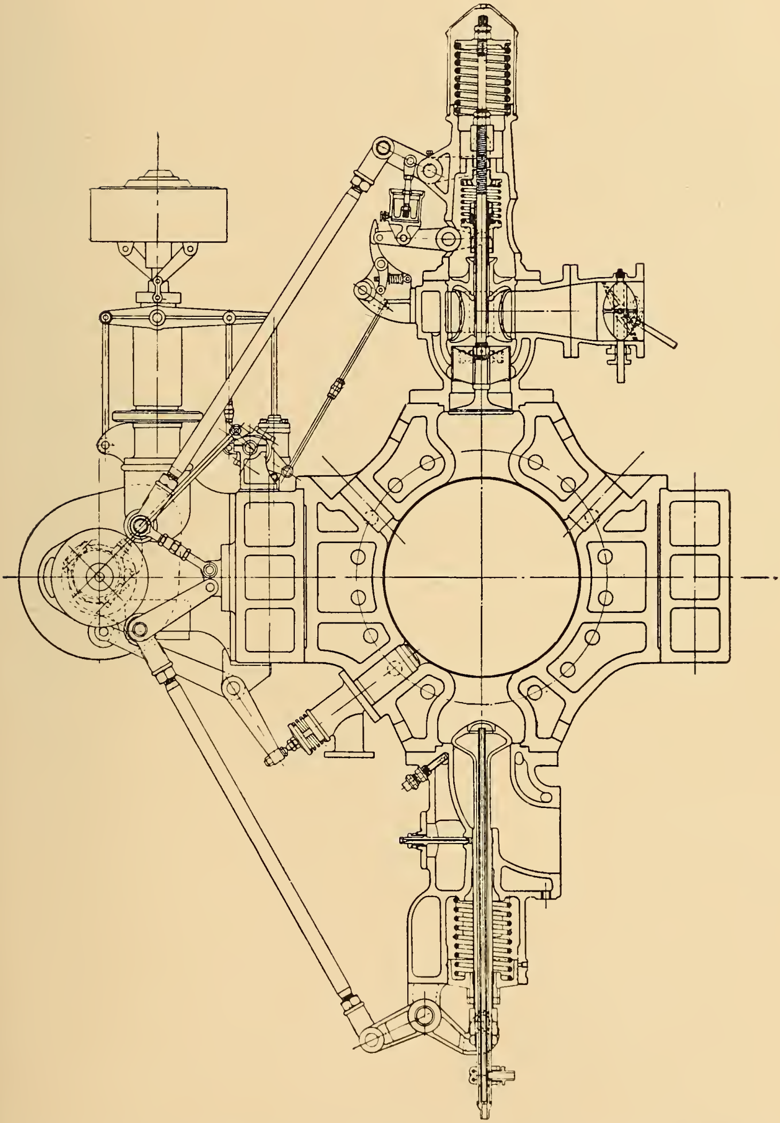


FIG. 33.—Arrangement of Valve Gear for Cockerill Engines (Richardsons, Westgarth and Co.). Note position of cam shaft for actuating inlet and outlet mushroom valves.

the more important types of large gas engines, the student may well turn his attention more closely to the **origin of the Clerk or two-stroke cycle.** That it is a very important cycle is obvious, bearing in mind the necessity of keeping down the capital cost of gas engines per horse-power. The Clerk cycle owes its origin to the inventiveness of Mr. Dugald Clerk, and the writer cannot do better than quote the inventor's remarks upon it before the Society of Arts in 1905 : "In the Clerk engine the motor cylinder had, at the front ends, large ports leading into an annular space, these being the exhaust ports. The compression space was conical, and the charge was sent in by means of a separate pump, which I called the displacer. The action of the engine was as follows : When the piston got to the out end of its stroke, and the crank was crossing the out centre, the piston overran the exhaust ports on the out-stroke, and covered them on the in-stroke. Meantime the pump or the displacer piston, which was attached to a crank at right angles in advance of the main crank, was sweeping in and giving its charge a slight compression. That charge passed through a connecting pipe, and through a check valve, into the conical end, displacing before it all the contents of the cylinder. When the main crank had returned about 40 degrees of its circle under the centre, these ports were closed. It opened about 40 degrees above and closed 40 degrees under, and in that time the displacer piston had gone fully in and discharged its charge into the cylinder and combustion space through the lift valve. Then the motor piston compressed the charge, and ignition took place at the in-end of the stroke, just as in the Otto cycle. The object of the invention was to enable one motor cylinder to give an impulse at every revolution. In the Otto cycle there is only one impulse for two revolutions, so far as the main cylinder is concerned. The Clerk engine gave one impulse for every revolution of the main crank in the main cylinder, but to make that possible it was necessary to provide an auxiliary crank and displacer cylinder. The idea, was, of course, to diminish the irregularity of the Otto cycle by having an impulse at every revolution, or more

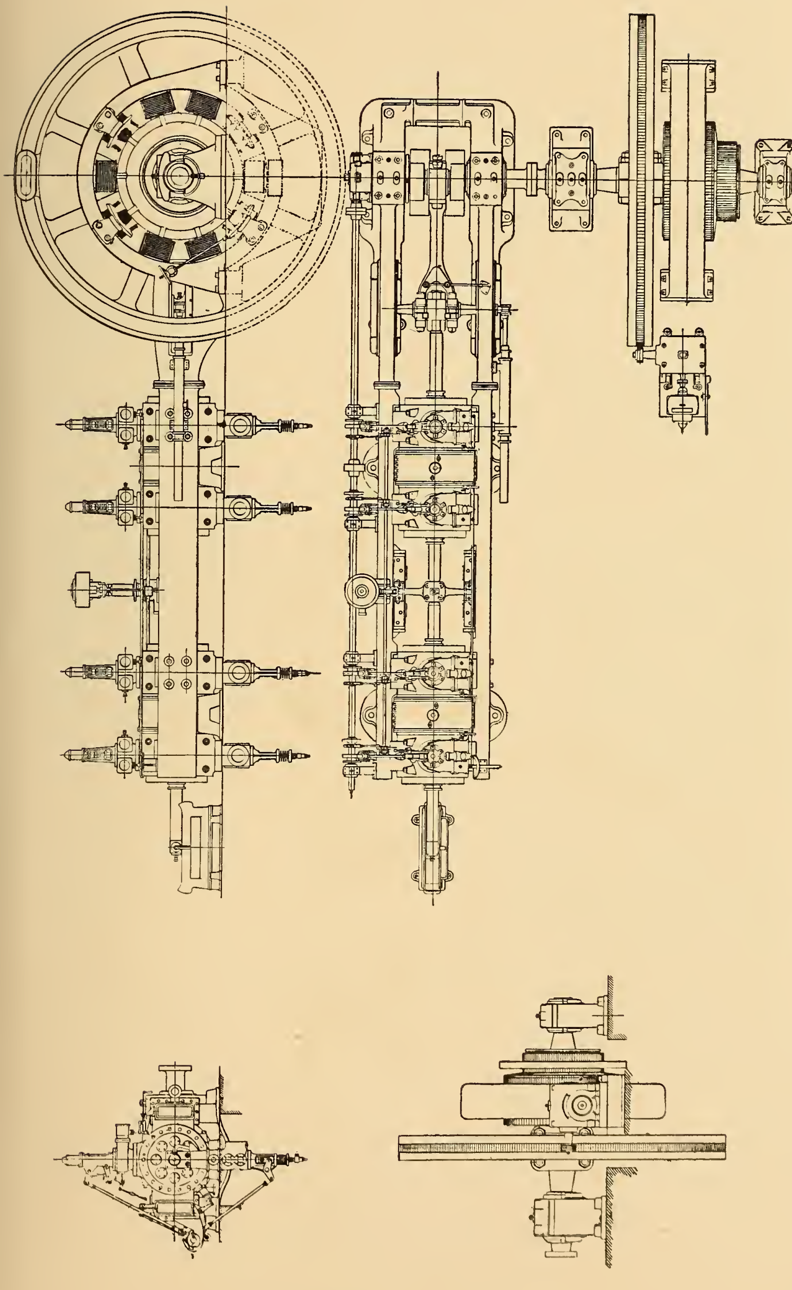


FIG. 34.—General arrangements of Cockerill Engine (Richardsons, Westgarth and Co.) for Dynamo driving. Part of a 4,500 H.P. plant for a shipyard in Hong Kong.

frequently, that is to say, two impulses per revolution, obtained by making the engine double-acting. The object was to get very much more power for a given weight of engine, as the pump was light and only required to deal with its charge at a low pressure. In construction the engine was a very simple one."

**65. Other Types of Engines. The Ehrhardt and Sehmer** engine works on the four-cycle double-acting principle, giving an explosion per revolution for each cylinder, so that for two cylinders placed tandem every stroke is a working stroke just as in a steam engine. With the tandem arrangement there is the advantage that the motion towards each dead centre is always preceded by a compression stroke; this leads to a cushioning action which is useful as an aid to overcoming the inertia effects due to the moving parts. These engines are very effectively water-cooled; the parts so treated include the pistons, piston rods, cones, glands, exhaust valves and valve casings. The engines are stated to be capable of driving alternators direct and allowing of parallel running. M. R. E. Mathot * writing in *Gas and Oil Power*, stated that: "A 600 h.p. double-acting engine, with two tandem cylinders, installed at the Kgl. Berginspection at Heintz, Saarbruck, Messrs. Ehrhardt and Sehmer, was tested at the end of last year, without cleaning, and after four months' continuous work with coke-oven gas of from 4,000 to 4,200 calories; the trial was carried out by the makers' engineers under the supervision of Kgl. Berginspection's engineer. Thanks to a large gasometer, the record of gas consumption was taken for a period of one hour. The mechanical efficiency was 83 per cent. The engine was new and was tested on a normal load. The dimensions of its pistons and piston rods were respectively 620 mm. and 170 mm., 750 mm. stroke, and 150 revolutions. It just reached 520 K.W., generating three-phase current. In the conditions of the trial the actual thermal efficiency was more than 31 per cent., or nearly 37½ per cent. of that indicated. Unfortunately, similar trials to this are rare because gasometers are not usually sufficiently large to measure the exact amount

* December 15, 1906.

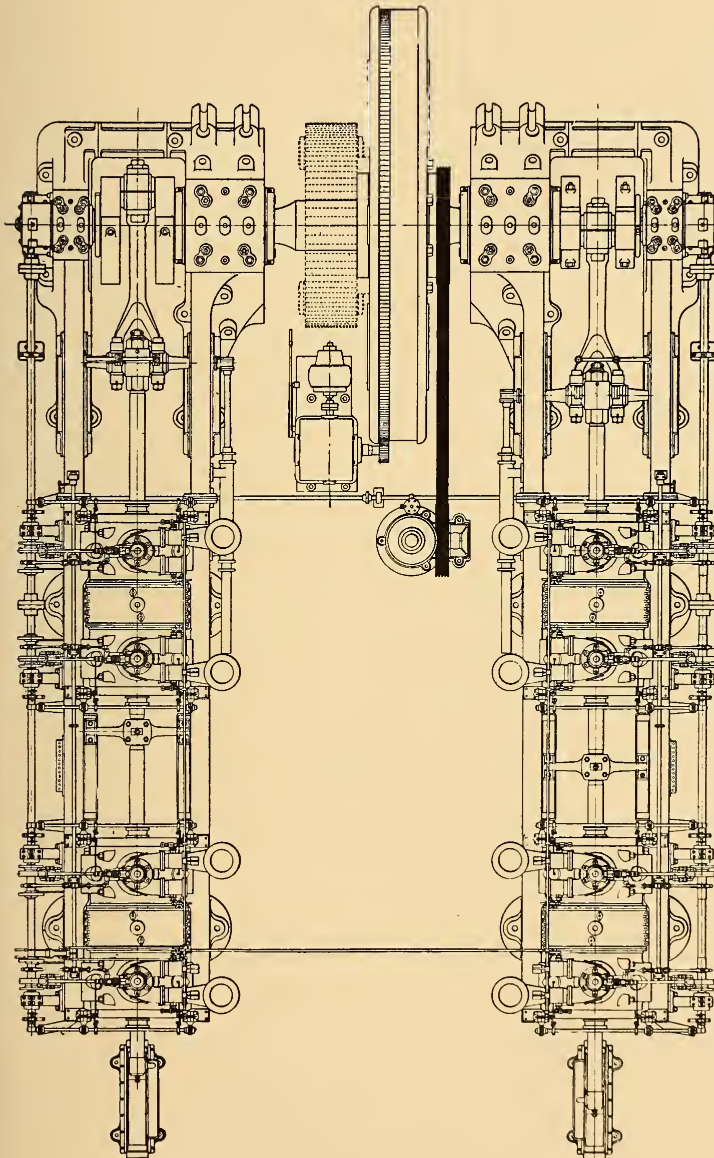


Fig. 35.—General arrangements of Cockerill Engine (Richardsons, Westgarth and Co.). Part of a 4,000 H.P. installation at the Kosmoid Tube Works, Dumbarton, to work on Producer Gas.

of gas used by the engine." Another well-known type is the **Premier gas engine.** Perhaps its most familiar feature is the scavenging of the exhaust products by means of an air blast. It has been stated that this blast is capable of keeping the cylinder interior almost entirely free from deposit when working with bituminous fuel gas plant. An account has been published * of a 1,200 h.p. four-cylinder gas engine by the Premier Gas Engine Co. which was constructed for direct coupling to a continuous current generator and consisted of two sets of tandem cylinders working on cranks set at 180 degrees apart. A four-stroke cycle was employed, so there were two working strokes per revolution. A scavenging charge of air supplied from a separate air cylinder at about 3 lb. per square inch was used to clear the cylinders of waste products. All the valves were placed on the cylinder covers; pistons and exhaust valves were both water-cooled. It was stated that, operating on producer gas, a compression pressure of 140 lb. per square inch could be used without any difficulty whatever from pre-ignition, and that a test on the engine showed the mechanical efficiency to be as much as 87 per cent. **The Westinghouse Co.** mainly produce a vertical engine, and are among the first to do so. Their 250 h.p. type has three cylinders and in general appearance resembles a high-speed vertical steam engine with its boxed-in crank chamber and splash lubrication. They also use the Otto cycle, and it is claimed that the governing is sufficiently steady to admit of the driving of alternators operated in parallel. The number of types of engines of less than 400 h.p. is very great, and the number of varieties increases as the output gets less than this. Of the smaller engines nearly all work on the Otto cycle and illustrations are shown of several types.

**66. Methods of Improving Efficiency** (Crossley and National).—The author has already endeavoured to make it clear that one of the chief causes why the efficiency of gas engines is not even higher than it is, lies in the high

* *Engineering*, January 11, 1907.

temperatures which occur during explosion and the very rapid rate at which heat is then abstracted by the walls. Two methods have been practically tried with a view to

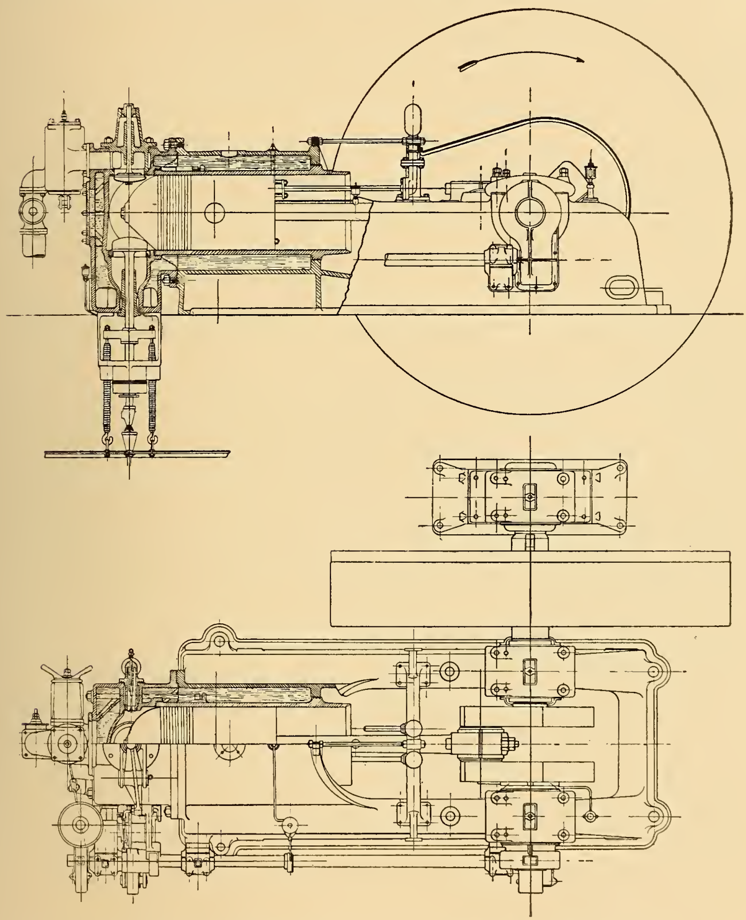


FIG. 36.—Standard Arrangement of Campbell Gas Engine. Note the disposition of the inlet and exhaust valves, and the water cooling arrangements. On the plan at the lower side is seen the half-time-shaft.

minimize this effect, the idea in each case being to reduce the maximum temperature of the cycle without, however, decreasing the mean pressure. These two are the water injection method of Messrs. Crossley Bros., and the super-

compression method of Mr. Dugald Clerk and the National Gas Engine Co.

**The Water Injection Method.**—Messrs. Crossley decided to try the effect of injecting a small spray of water into the cylinder during the suction stroke. The water, entering as a fine spray in part of the air supply was as evenly distributed as possible and did not form a water film on

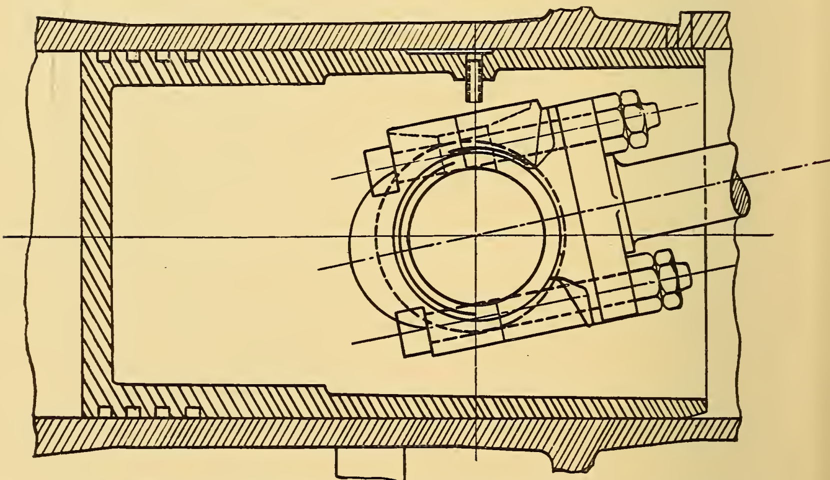


FIG. 37.—Typical Gas Engine Piston and small end of Connecting Rod. Note method of lubricating small end.

the cylinder walls. Very little water is required, because of the high value of its latent heat. As the mixture explodes the water mist is evaporated into steam and the heat so absorbed prevents the temperature of the gases from rising unduly high. A 50 h.p. engine so adapted was tested by Professor Burstall in 1904 and the following records taken :—

| | |
|---|---|
| Size of engine . . . | 14 in. ×21 in. |
| Duration of test . . . | 6 h. 45 m. |
| Average revs./min., . . . | 166·02 |
| ,, explosions/min. . . | 81·2 |
| ,, mean pressure lb./in.² . | 91·44 |
| ,, i.h.p. . . . . | 60·5 |
| ,, b.h.p. . . . . | 49·7 |
| Mechanical efficiency . . . | 82·2 per cent. |

| | |
|---|---|
| Gas used per i.h.p.-hour . . | 11·77 cu. ft. |
| ,, ,, b.h.p.-hour . . | 14·43 ,, |
| Calorific value of gas (lower value) | 578 B.T.U. per cu. ft. |
| Thermal efficiency on i.h.p. . | 37·43 per cent. |
| ,, ,, ,, b.h.p. . | 30·8 ,, |
| Water used in cylinder . . | 0·131 lb. per minute. |
| ,, discharged from jacket | 25·66 ,, ,, |
| Average rise in temperature of jacket water . . . . | 77·52° F. |
| Mean temperature of exhaust as measured by Callendar pyrometer . . . . . | 718° F. |

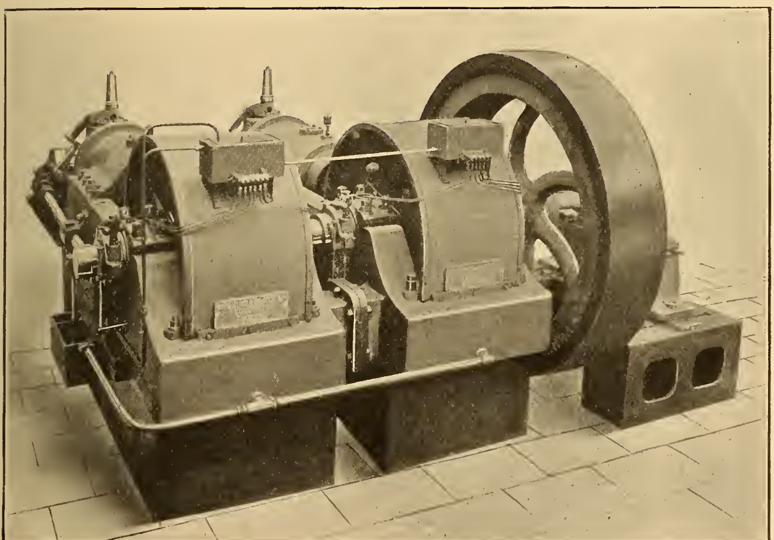


Fig. 38.—Fielding Gas Engine. 290 B.H.P. Two-cylinder-side-by-side type. Note the lubricating arrangements.

The ratio of air to gas was 10·2 and the compression ratio 8·7 (obtained by dividing the clearance volume of 0·243 cu. ft. into the cylinder volume of 1·872 cu. ft.).

Now a compression ratio of 8·7 corresponds on the "air standard" to an efficiency of $1-\left(\frac{1}{8\cdot7}\right)^{0\cdot4}$ which equals 0·58, and as the actual efficiency found was 0·37 it follows

that the engine achieved nearly 64 per cent. of the "air standard" efficiency. This is a higher ratio than any of those given by Mr. Dugald Clerk in his 1907 paper before the Institution of Civil Engineers ("On the Limits of Thermal Efficiency in Internal-Combustion Motors"), which showed no higher ratio than 59 per cent. and that only in the case of a maximum temperature of 1,098° C., whereas when the temperature rose to 1,750° the ratio fell to 50 per cent. and below. On this method of comparison, therefore, the water injection method shows to advantage.

**The Super-Compression Method.**—This method is due to the ingenuity of Mr. Dugald Clerk, who in his James Forrest Lecture (1904) before the Institution of Civil Engineers, described it thus: "Some time ago it appeared to me possible to reduce maximum temperatures by increasing the charge-weight per stroke given to an engine. I had experimented with two engines, one having a 7 in. cylinder, 15 in. stroke, and the other a 10 in. cylinder, 18 in. stroke. These engines, which are of the ordinary standard four-cycle type, are allowed to take in the usual charge of gas and air; then at the end of the stroke a further charge of air or other inert fluid is added to increase the pressure in the cylinder to 7 lb. or 8 lb. per square inch above atmosphere before the return of the piston. A small part of the return stroke is, however, made before the pressure can be materially increased as the added charge takes some time to fill the cylinder. This has the effect of increasing the charge weight present in the cylinder by about 40 per cent. and of increasing the pressure of compression without, however, increasing the temperature of compression. Indeed in both experiments the temperature of compression was diminished. As the charge present is constant so far as gas is concerned the maximum temperature capable of being produced is much reduced. The maximum temperature shown by the diagrams taken by me from these two engines is about 1,200° C. Experiments were made and it was found that the heat-flow was reduced to about two-thirds, and further that the mean available pressure was increased about 20 per cent."

The thermal efficiency of an engine which on working without super-compression was 27·7 per cent. showed an increase to 34·4 per cent. when super-compression was

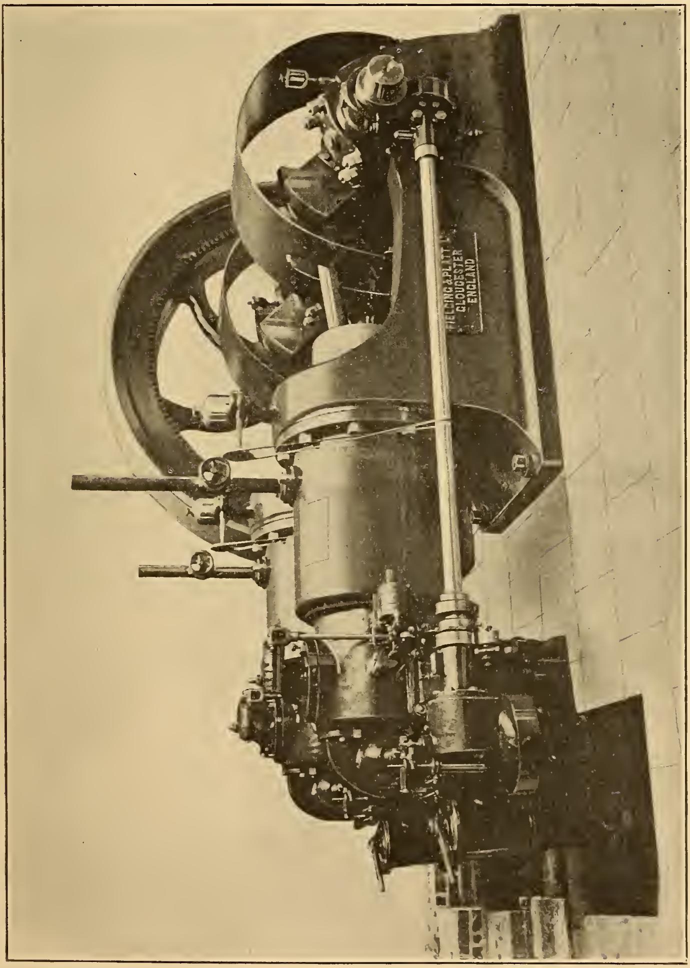


Fig. 39.—130 B.H.P. Fielding Gas Engine. Two-cylinder-side-by-side type.

adopted. One sees therefore that if the atmospheric pressure were 50 per cent. higher than it is, it would suit the working of gas engines a great deal better.

The increases in thermal efficiency obtained by the water

injection and the super-compression methods are of course desirable in themselves, but they are really the most welcome for what they bring in their train, viz. freedom from cracking of cylinders and pistons. Low efficiency means a large amount of heat being passed away through the walls to the cooling water, and the larger the engine the larger the amount of heat to be got rid of in this way and the smaller in proportion to cubic contents does the cooling surface become. This means a steep heat gradient in the metal, and this in turn leads to the failure of engines owing to the cracking of ends or walls or sometimes of pistons themselves if water cooled as is usual in the larger sizes. Manufacturers may therefore be said to be seeking high efficiencies not for the resulting economy in fuel (for the gas engine has there long left all competitors behind) but on account of the increased freedom from mechanical difficulties in operation. It is a particularly happy feature of the case that gas engine builders seeking and obtaining increased reliability of operation should also find that the same improvement leads also to greater fuel economy. Economy in fuel is of no great direct importance in the case of large engines working on blast furnace or coke-oven gases, but in other cases it is often an advantage. Improved methods which allow of the maximum cyclic temperature being reduced without any loss of power can also be pressed in the direction of increasing the mean pressure considerably without, however, raising the temperature so high as it was previously. This leads to greater output, but the pressure at exhaust is considerable, and it would in such cases be an advantage to use this exhaust in another cylinder and so compound the engine. Efforts in this direction are being made.

**67. The Indicator.**—A very important instrument used in connexion with gas engines is the indicator. It is an apparatus which when attached to an engine draws a curve showing how the pressure in the cylinder varies at different points in the stroke. The best known modern form is the Crosby shown in Fig. 41. On the left of the illustration will be seen a small cylinder containing a cup-shaped piston which is regulated in its upward motion by the down-

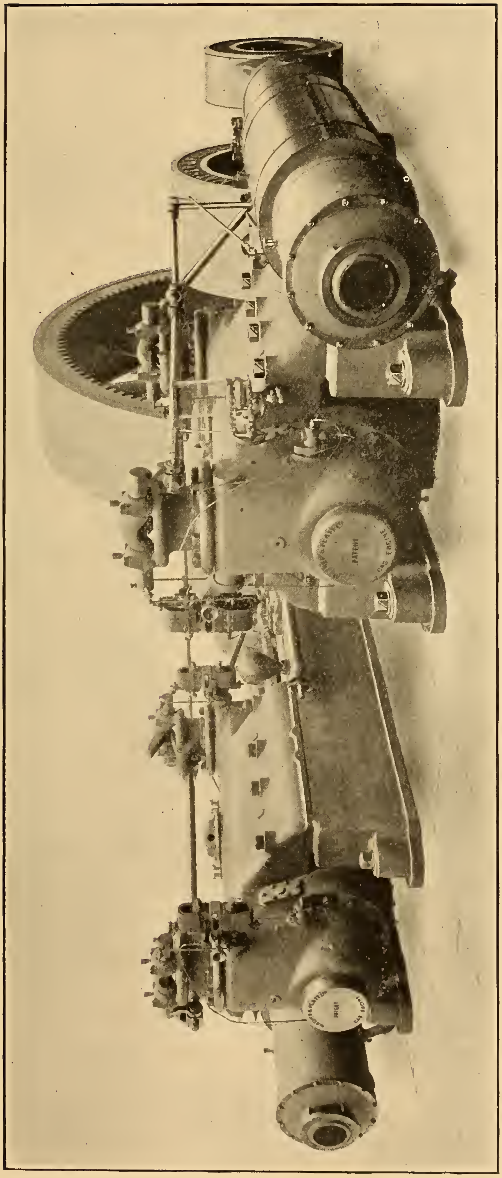


FIG. 40.—700 B.H.P. Koerting Gas Engine with grooved flywheel for rope-driving (Mather and Platt). Note the ignition arrangements.

ward push of the strong spring seen above. When the indicator is screwed on to the engine cylinder the pressure causes the indicator piston to rise through a distance proportional to the force exerted. The little piston rod rises also and communicates its motion to the long sloping lever seen above. This lever carries at its far end a pencil which traces a line on a paper sheet fastened round the drum seen on the right which is made to oscillate to and fro by the

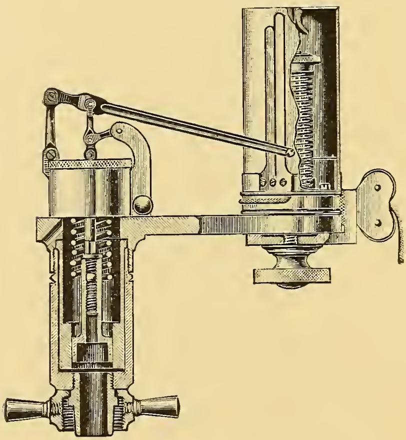


Fig. 41.—Crosby Indicator with Internal Spring.

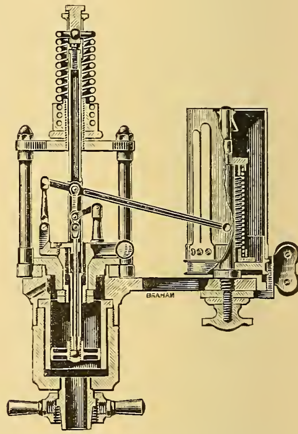



Fig. 42.—Crosby Indicator with External Spring.

cord shown on the extreme right of the diagram being attached to some part which has a motion similar to that of the piston, but less in amount. The pencil therefore traces out the closed curve known as indicator diagram. These diagrams will be familiar to all readers of this book and probably the indicator also—if not it would be well worth any student's time to spend some hours examining a Crosby instrument and taking readings with it.

Fig. 42 shows another form of the instrument having the *spring outside* where it is less affected by the heat and so gives a better reading. Of course all these springs are carefully calibrated first so that it is known how much pressure is represented by a rise of the pencil point equal to, say, 1 inch. There are certain qualities which a well-

designed indicator should have. It must have a spring stiff enough to ensure that the maximum pressure will come well within its range. It must have a well-designed piston, as light as is consistent with strength, which will move freely in the cylinder. A slight leakage of steam is much less of an evil than any chance of the piston sticking or jambing. In the Crosby form the piston is made from a solid piece of tool steel, hardened and then ground and lapped to gauge. It is provided with a socket to receive the bead at the end of the spring and has screw adjustments for locking the spring in place. The rod is made hollow for lightness and is threaded at its lower end for attachment to the piston. The spring is made with a double coil so as to centre it the better. It has already been mentioned that the to and fro motion of the paper is obtained from a cord attached at its other end to some point in the upper part of a swinging lever of which the lowest point is connected with the engine piston or some part that moves with it so that the motion of the engine piston is reduced to a convenient degree. There is also a Crosby reducing device for doing this in a simpler way. It is shown illustrated in Fig. 43 and its principle of action is easily seen therefrom. In this device the cord at the bottom can be fastened direct to the crosshead, or other part attached to the piston, the cord passing over guide pulleys if necessary. It is better, however, not to have a longer cord than necessary as its stretching with the pull put on it may introduce serious error into the indicator card.

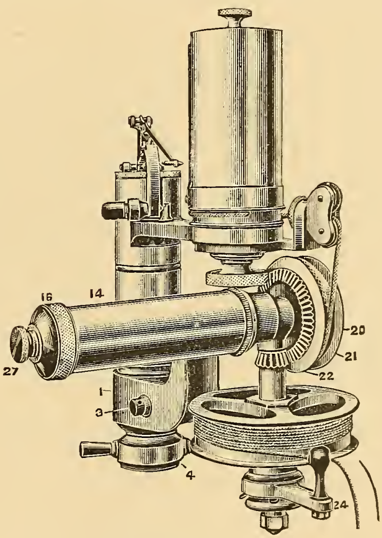



Fig. 43.—Reducing Gear for Crosby Indicator.

For very accurate work the cord has sometimes been replaced by steel wire.

**68. Reflecting Types of Indicator.** Although as has been stated the indicator is a very important instrument in gas engine work, it does not nevertheless occupy the important position it does in steam engine practice where lower maximum pressures are met with. This may

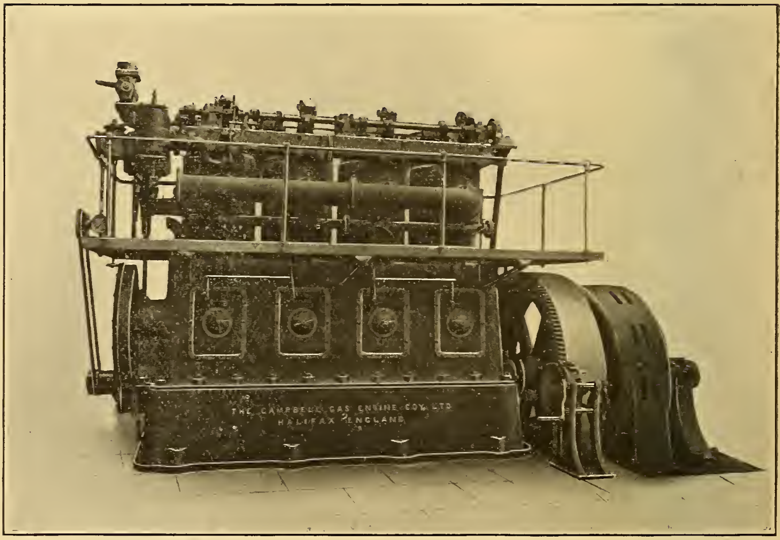



FIG. 44.—360 B.H.P. Vertical Four-cylinder Campbell Gas Engine, coupled to Electric Generator. Note position of cam shaft.

be due to the fact that most of the steam engine work was done at a time when the unavoidable errors of the indicator instrument were less well realized. Latterly it has become common to record the motion of the indicator piston by means of a beam of light reflected, as in a reflecting galvanometer, from a vertical mirror which is caused to tilt as the piston rises ; at the same time the frame in which the mirror is held is made to move angularly to and fro in time with the motion of the crosshead, thus producing by the combination of motion the familiar shape of the indicator card. The beam of light, unlike the steel levers of the older form of indicator, has no weight and therefore

no inertia to make it lag behind its true position. Professor Hopkinson claims that with an instrument devised by him on the reflecting principle, the indicated horse-power can be measured with an error of less than 1 per cent., whereas in the older forms errors of 5 per cent. or more were common. Professor Hopkinson * in a paper read by him recently made the following comment :—

"In the report of the Committee of the Institution of

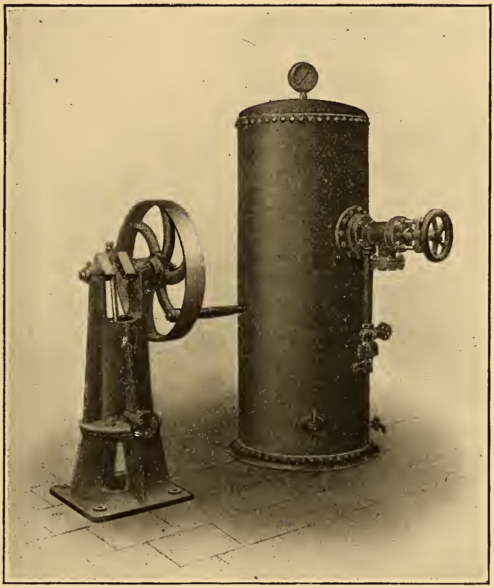


FIG. 45.—Air Pressure Vessel and Pumps for starting Campbell Gas Engine.

Civil Engineers on the Efficiency of Internal Combustion Engines the following remark occurs : 'It would be desirable but for one circumstance to calculate the relative efficiency only from the indicator horse-power. But it appears that in the case of gas engines, and especially gas engines governed by hit-or-miss governors, the indicator diagrams do not give as accurate results as is generally supposed.

* "On the Indicated Power and Mechanical Efficiency of the Gas Engine," Institute of Mechanical Engineers, 1907.

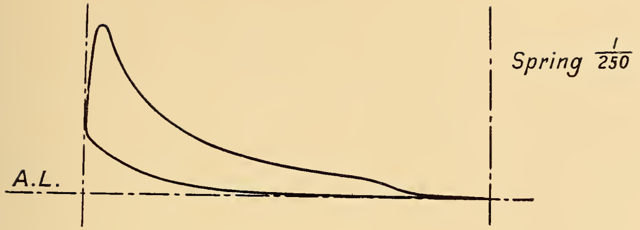
The diagrams vary much more than those of a steam engine with a steady load, and the mean indicated horse-power, from the diagrams taken in a trial, may, it appears, differ a good deal from the real mean power.' This statement is fully borne out by the tests of the Committee, which show that the mechanical efficiency taken as the ratio of brake to indicated power varied from 80 per cent. to 94 per cent. in the three engines tested. These engines were of similar type, but of different sizes, and whereas the smallest of 5 h.p. showed a mechanical efficiency of 90 per cent., the intermediate engine of 20 h.p. showed a lower efficiency of 80 per cent. The Committee remarked that these values were obviously incorrect, and the values adopted by them for the mechanical efficiency were obtained by running the engine light and making an estimate of the indicated horse-power under these conditions. Assuming that the mechanical loss is constant at all loads, the indicated power at full load can be determined by adding the power absorbed at no load to the brake-power. The mechanical efficiencies of the three engines found in this way were respectively :

| | | | |
|---|---|---|---|
| Engine . . . . | *L* | *R* | *X* |
| Mechanical efficiency . | 0·86 | 0·866 | 0·888 |

"These results are just what would be expected; the mechanical efficiency showing a slight improvement with the size of the engine."

**69.** The method of getting the i.h.p. by assuming that the mechanical loss is constant at all loads is hardly satisfactory. It is obvious that to assume that the friction at, say, the big and small ends of the connecting rod, or on the piston, will be the same whether there is any thrust in that rod or not cannot be strictly accurate, and even if the result of making this assumption is to produce a result reasonable in itself, that is not a valid reason for accepting it.

Professor Hopkinson decided to test the truth of this assumption for himself, and he carried out a complete series of tests, using the reflecting type of indicator already mentioned. He found that "the difference between indicated horse-power and brake horse-power is rather less than the



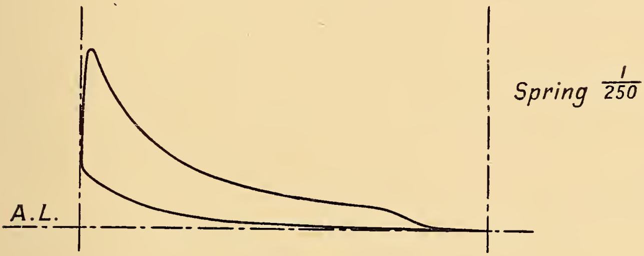




FIG. 46.—Indicator Diagrams taken from 500 B.H.P. "Oechelhäuser" Gas Engine. Cylinder 26 $\frac{9}{16}''$ bore. Front stroke $37\frac{1}{2}''$. Back stroke $37\frac{1}{2}''$. When calculating I.H.P. from these cards it must be remembered that the engine is a two cycle one. Compare with Fig. 50.

horse-power at no load under the same conditions of lubrication, mainly because of the difference in the power

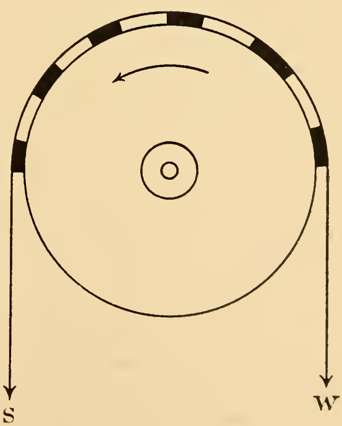



FIG. 47.—B.H.P. tests. When the mechanical efficiency of an engine has to be measured, both I.H.P. and B.H.P. must be found. The B.H.P. when measured by a friction brake as shown in this diagram

$$= \frac{W - S}{33000} \times c \times \text{r.p.m.}$$

where $c$ is circumference of pulley in feet, and W and S are measured in pounds.

absorbed in pumping. In the particular engine tested, the error from this cause in obtaining the indicated power would amount to about 5 per cent. The friction is substantially constant from no load to full load, provided that the temperature of the cylinder walls is kept the same, but the influence of temperature is very great." He found the mechanical losses in a 41 h.p. engine to be as follows :—

| | | | |
|---|---|---|---|
| Suction . . . . . | 3·4 | per cent. of i.h.p. | |
| Piston friction. . . . | 6·1 | ,, | ,, |
| Other friction (valve lifting, etc.) | 2·7 | ,, | ,, |
| Total . . . . | 12·2 | ,, | ,, |

The various efficiency figures for this engine were :—

| | |
|---|---|
| Thermal efficiency . . | $33\frac{1}{2}$ to 37 per cent., according to strength of mixture. |
| Mechanical efficiency for medium charge . . | 85 to 90 per cent., according to jacket temperature. |
| "Air-standard" efficiency. | 52·2 per cent. (on compression ratio of 6·37). |
| Efficiency relative to "air-standard" . . . | 0·64 to 0·71. |

**70. Analysis of Motion of Indicator Piston.**—The piston used in the indicator instrument cannot be absolutely weightless whatever improvement may be made in reducing the weights of the moving levers (either by adopting lighter scantlings or by using a beam of light). Let the pressure acting on the base of the piston of mass $M$ at any time to be $p$, also let piston area $= a$ and the motion of the piston be $S_2$ inches for each pound per sq. inch of pressure acting upon it. Then the forces acting on the piston when at a point $x$ above its lowest position are :—

upwards $\qquad p \times a$

downwards $\qquad \frac{x}{S_2} \cdot a + M \cdot \frac{d^2x}{dt^2}$

therefore $\qquad M\frac{d^2x}{dt^2} + \frac{a}{S_2} \cdot x = p.a.$

or
$$\frac{d^2x}{dt^2} + \frac{a}{S_2M} \cdot x = \frac{pa}{M} \quad \ldots \ldots (1)$$

Integrate this. The Particular Integral is

$$x = \frac{1}{D^2 + \frac{a}{S_2M}} \cdot \frac{pa}{M} = \frac{S_2M}{a} \cdot \frac{1}{1 + \frac{S_2M}{a}D^2} \cdot \frac{pa}{M}$$

so that $x = \frac{S_2 M}{a} \times \frac{pa}{M} = p.S_2$

and the Complementary Function is

$$x = A \sin \sqrt{\frac{a}{S_2 M}} \,.\, t + B \cos \sqrt{\frac{a}{S_2 M}} \,.\, t$$

So that the Complete Integral is

$$x = A \sin \sqrt{\frac{a}{S_2 M}} \,.\, t + B . \cos \sqrt{\frac{a}{S_2 M}} \,.\, t + p.S_2.$$

Now when $t = o, x = o$

therefore $B = -pS_2$

$$\frac{dx}{dt} = \sqrt{\frac{a}{S_2 M}} \,.\, A \cos \sqrt{\frac{a}{S_2 M}} \,.\, t - B \sqrt{\frac{a}{S_2 M}} \,.\, \sin \sqrt{\frac{a}{S_2 M}} \,.\, t$$

and when $t = o, \frac{dx}{dt} = o$

therefore $A = o$

Substitute these values of $A$ and $B$ and

$$x = pS_2 \left\{ 1 - \cos \sqrt{\frac{a}{S_2 M}} \,.\, t \right\}.$$ This means that the piston rises to a height $pS_2$ and then oscillates about that position with a frequency equal to $\frac{1}{2\pi} \sqrt{\frac{a}{S_2 M}}$.

All this assumes, however, that $p$ is a constant or that it increases with such rapidity that it assumes its final value before the indicator piston has had time to move. It would have been more accurate to assume $p$ to rise from zero to its final value in, say, $\frac{1}{n}$ th part of a second and to consider what happens during this interval. To do this put $p = a_1 . t$ where $a_1$ has the constant value given by the equation :—final value of pressure $= \frac{a_1}{n}$.

Equation (1) now becomes

$$\frac{d^2 x}{dt^2} + \frac{a}{S_2 M} \cdot x = \frac{a}{M} \cdot a_1 t \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad (2)$$

and the Particular Integral

$$x = \frac{S_2 M}{a} \left( 1 + \frac{S_2 M}{a} D^2 \right)^{-1} \cdot \frac{a a_1}{M} \cdot t$$

$$= \frac{S_2 M}{a} \cdot \frac{a a_1}{M} \cdot t = S_2 a_1 t.$$

Therefore the Complete Integral would be

$$x = A \sin \sqrt{\frac{a}{S_2 M}} \,.\, t + B \cos \sqrt{\frac{a}{S_2 M}} \,.\, t + S_2 a_1 t.$$

And since $x = o$ when $t = o$

therefore $B = o$

Again $\frac{dx}{dt} = \sqrt{\frac{a}{S_2M}} \cdot A \cos\sqrt{\frac{a}{S_2M}} \cdot t + S_2a_1$

but when $x=o$ $t=o$

so that $$o = \sqrt{\frac{a}{S_2M}} \cdot A + S_2a_1$$

and $$A = -S_2a_1\sqrt{\frac{S_2M}{a}}$$

This gives us $$x = S_2a_1t - S_2a_1\sqrt{\frac{S_2M}{a}} \sin\sqrt{\frac{a}{S_2M}} \cdot t$$

$$= S_2a_1\left\{t - \sqrt{\frac{S_2M}{a}} \cdot \sin\sqrt{\frac{a}{S_2M}} \cdot t\right\}$$

Now $S_2a_1t$ is height to which piston would rise under the slow static pressure—call it $h$ so that $S_2a_1t = h$, and let $f$ be the frequency of the free vibration of the indicator piston. Then

$$f = \frac{1}{2\pi}\sqrt{\frac{a}{S_2M}} \quad \text{or} \quad 2\pi f = \sqrt{\frac{a}{S_2M}}$$

so that $$x = h - \frac{S_2a_1 \cdot}{2\pi f} \sin\sqrt{\frac{a}{S_2M}} \cdot t$$

$$= h - \frac{h}{2\pi ft} \cdot \sin 2\pi ft$$

or $$x = h\left\{1 - \frac{1}{2\pi ft} \cdot \sin 2\pi ft\right\} \quad \ldots \quad (3)$$

This means a fractional lag of $\frac{1}{2\pi ft}$ as a maximum, but for any particular case it can be calculated thus. We may put $f$ as 300, which about represents the best modern practice using an instrument of the Hopkinson type.

Then

$$x = h\left\{1 - \frac{1}{1890t} \sin 1890t\right\}$$

It will be useful to compute a few values for this for cases in which the value of $t$ is much shorter than the periodic time of the instrument. When this is so sin $1890t$ can be written with sufficient accuracy as

$$(1890t) - \frac{(1890t)^3}{6}$$

or $$x = h\left\{1 - 1 + \frac{(1890t)^2}{6}\right\}$$

$$= h \cdot \frac{(1890t)^2}{6} = 600{,}000ht^2.$$

The relation between $x$ and $t$ in these early stages is therefore parabolic. The time $t$ starts, so to speak, first, but $x$ soon increases and gradually catches up.

Thus for $t = \frac{1}{100000}$ sec.

$$\frac{x}{h} = \frac{600{,}000}{(100{,}000)^2} = 0{\cdot}00006$$

For $t = \frac{1}{10000}$ sec.

$$\frac{x}{h} = 0{\cdot}006$$

for $t = \frac{1}{1000}$ sec.

$$\frac{x}{h} = 0{\cdot}6$$

but for this value of $t$ our approximation no longer holds. For $t = \frac{1}{1000}$ the calculation should proceed thus

$$\frac{x}{h} = \left(1 - \frac{1000}{1890} . \sin 1{\cdot}890\right)$$

$$= 1 - \frac{1}{1{\cdot}89} \sin 108°$$

$$= 1 - \frac{1}{1{\cdot}89} \times 0{\cdot}95 = 1 - 0{\cdot}50$$

$$\therefore \frac{x}{h} = 0{\cdot}50$$

showing that the instrument is picking up. Evidently therefore it will not do to use an instrument for recording an explosion occurring in $\frac{1}{1000}$ sec. unless its own frequency exceeds 300.

The following table shows a series of values, and in Fig. 47 they are shown plotted.

| $t_{\text{secs.}}$ | $1890t$ | $\sin 1890t$ | $\left(\frac{1}{1890t} . \sin 1890t\right)$ | $\frac{x}{h}$ |
|---|---|---|---|---|
| $\frac{1}{100000}$ | — | — | — | 0·00006 |
| $\frac{1}{10000}$ | — | — | — | 0·006 |
| $\frac{1}{2000}$ | — | — | — | 0·15 |
| $\frac{1}{1000}$ | 1·89 | 0·95 | 0·50 | 0·50 |
| $\frac{1}{500}$ | 3·78 | —0·60 | 0·16 | 1·16 |
| $\frac{1}{400}$ | 4·72 | —1·00 | —0·21 | 1·21 |
| $\frac{1}{250}$ | 7·56 | 0·96 | 0·13 | 0·87 |
| $\frac{1}{100}$ | 18·9 | 0 | 0 | 1·00 |

Whenever $t$ is a multiple of $\frac{1}{600}$ the value of $\frac{x}{h}$ will be 1·00. The above curve does not of course take account of the frictional forces which prevent the indicator piston continuing to vibrate indefinitely. Students are recommended to work the problem out, introducing into equation (2) a term representing the frictional force. The result will be to multiply the oscillatory term by a factor of the type $\epsilon^{-qt}$ which, when the student has plotted the resulting curves, will show that the straight line is soon followed once the curve comes up to and crosses it. From the curve in Fig. 48 it is clear that

for recording an explosion occurring in $\frac{1}{10000}$ sec. this indicato: with its $\frac{1}{300}$ period would be inadequate. The piston would scarcely have moved. For an explosion occupying ten times as long, i.e. $\frac{1}{1000}$ sec., the indicator would still be lagging a long way behind. For a $\frac{1}{500}$ sec. explosion the actual maximum pressure would be very fairly represented, but not the shape of the explosion wave. In fact for useful readings the instrument should not be used for any sharper explosion than $\frac{1}{200}$ sec. For an ordinary gas engine explosion occurring in $\frac{1}{100}$ sec. the instrument would be quite satisfactory.

**71. Heat Balance-sheets.**—A heat balance-sheet as applied

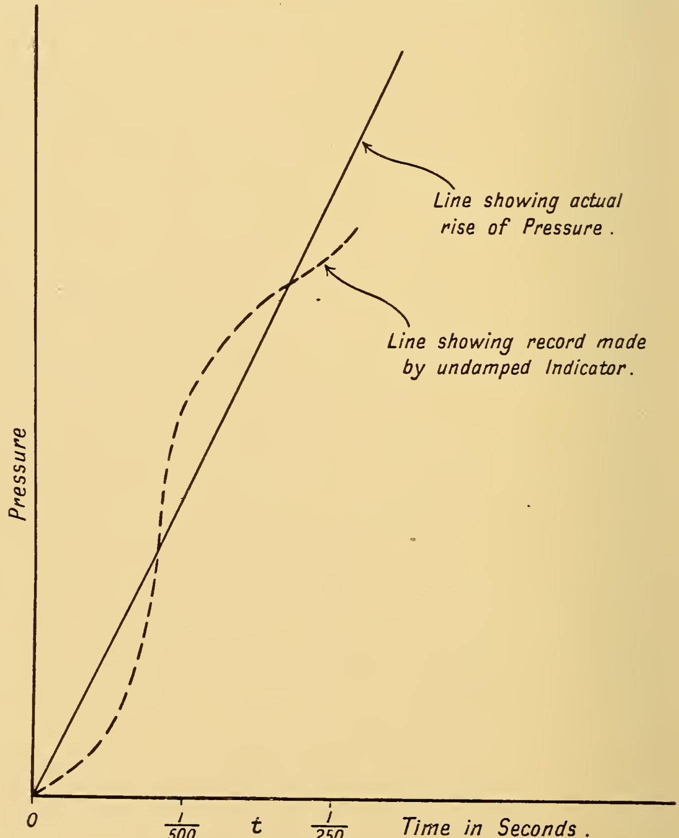



FIG. 48.—Diagram illustrating the way in which an undamped Indicator would follow a rapid explosion. Period of Indicator, $\frac{1}{300}$ sec.

to a gas engine is a statement of the way in which the total amount of heat passed into the engine is employed. In the

early days of gas engine work it was easy to remember that roughly—

| | |
|---|---|
| Heat passed to water jacket . . | 40 per cent. |
| Heat left in exhaust gases . . | 40 ,, |
| Heat converted into work . . | 20 ,, |
| | 100 ,, |

Such balance-sheets have, however, lately become a good deal more complicated. First, there is the difficulty of knowing how much heat the exhaust gases really carry away, the specific heat not being accurately known; and then there is the difficulty that the exhaust gases on their way out usually part with some of their heat to the water jacket. This leads to part of the loss being counted twice over so that the total instead of coming out as 100 per cent. often comes out as 101 per cent. or 102 per cent.

In the experiments made by the Institution of Civil Engineers Committee already referred to the full-load heat balance-sheet was given as :—

| Designation of Engine. | *L.* | *R.* | *X.* |
|---|---|---|---|
| Exhaust waste . . . . . . | 35·3 | 40·0 | 39·5 |
| Jacket waste . . . . . . | 23·5 | 29·3 | 25·0 |
| Radiation . . . . . . . | 7·6 | 10·0 | 7·3 |
| B.H.P. . . . . . . . . | 26·7 | 28·3 | 29·8 |
| Total . . . . . . . | 93·1 | 107·6 | 101·6 |

In these experiments the exhaust waste was measured by passing the exhaust gases into a water-jet calorimeter. Jacket waste was measured as the product of quantity of cooling water passed and rise of temperature. Radiation includes engine friction as well as radiation proper. B.H.P. was measured by a rope brake.

Engine *L* shows a deficit in the total, so that there must have been something wrong in the experiments. Mr. Dugald Clerk in his paper * before the Institution of Civil

* Read February 26, 1907.

Engineers, "On the Limits of Thermal Efficiency in Internal Combustion Motors," endeavoured to correct this measurement from several different possible points of view. He also extended the same treatment to tests *R* and *X* in order to get the true balance-sheet, and putting in i.h.p. instead of b.h.p. (the Committee's records were complete enough to permit of this), he found :—

| Designation of Engine. | *L.* | *R.* | *X.* |
|---|---|---|---|
| Exhaust waste . . . . . . | 41·0 | 37·1 | 39·9 |
| Jacket waste and radiation . | 27·2 | 29·6 | 25·4 |
| I.H.P. . . . . . . . . . | 31·8 | 33·3 | 34·7 |
| Total . . . . . . . | 100·0 | 100·0 | 100·0 |

Mr. Clerk then points out that the 27·2 per cent. of jacket waste and radiation for test *L* is obviously too low, and that heat appears to have been lost in some way. He therefore took the total of the exhaust waste and jacket waste and radiation items, i.e. 68·2 per cent. and attributed 34·1 per cent. to each, so making the balance sheet into :—

| Designation of Engine. | *L.* | *R.* | *X.* |
|---|---|---|---|
| Exhaust waste . . . . . . | 34·1 | 37·1 | 39·9 |
| Jacket waste and radiation . | 34·1 | 29·6 | 25·4 |
| I.H.P. . . . . . . . . . | 31·8 | 33·3 | 34·7 |
| Total . . . . . . . | 100·0 | 100·0 | 100·0 |

Mr. Clerk considered this balance-sheet probably represented the distribution of heat in the engines more accurately than either of the others.

**72.** These various attempts at a heat balance-sheet have been given in order to show how very difficult it is to obtain a really accurate statement. The exhaust wastes originally given for *L*, *R* and *X* were 35·3 per cent., 40·0 per

cent. and 39·5 per cent., and have now become 34·1 per cent., 37·1 per cent. and 39·9 per cent.

But the matter does not end even here as Mr. Clerk

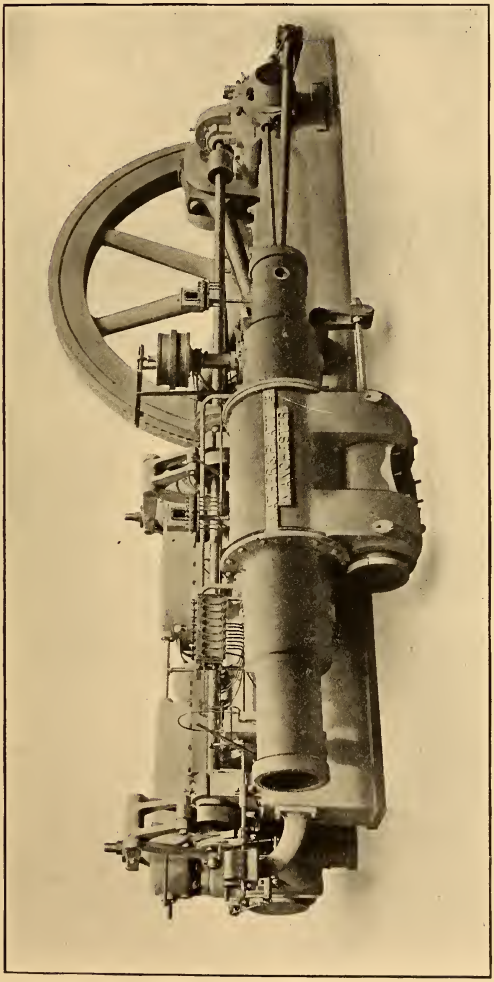


FIG. 49.—400 B.H.P. Koerting Engine (Mather and Platt). One of the two engines at the Shelton Iron and Steel Co.'s Works. Note position of cam shaft.

brought into use the values found by him for the specific heat—values which showed a marked increase with rise of specific heat—and used them in some separate experiments

of his own with the engine $X$ used by the Committee. He then found that the balance-sheet became :—

| | | |
|---|---|---|
| Heat-flow during explosion and expansion . . . . | 16·1 | per cent. |
| Heat contained in gases at end of expansion . . . . | 49·3 | ,, |
| I.H.P. . . . . . . | 34·6 | ,, |
| | 100·0 | |

Compare this with the balance-sheet given on p. 150 based on the Committee's experiments :—

| | Committee's Trials. | Mr. D. C's. Trials. |
|---|---|---|
| Heat-flow during explosion and expansion | 25·4 | 16·1 |
| Heat contained in gases at end of expansion | 39·9 | 49·3 |
| I.H.P. . . . . . . . . . . . . . | 34·7 | 34·6 |
| Total . . . . . . . . . . | 100·0 | 100·0 |

The discrepancies shown here are indeed serious. Mr. Clerk's comment on them is as follows : "The indicated work is practically the same in both trials and the sum of the other two items is the same also, but the distribution is different. Less heat flows through the cylinder-walls as determined by the author's (Mr. Clerk's) new method, and the exhaust gases contain more heat than the Committee's calorimeter trials show. The ordinary trials show 9·3 per cent. too much heat as passing through the cylinder-walls, and practically the same amount too little appears in the exhaust calorimeter. That is, 18·8 per cent. of the total heat remaining in the hot gases at the end of the expansion passes into the cylinder water-jacket during the flow through the exhaust valve upon the first opening and while the piston is making its exhaust stroke. This seems to be a quite reasonable portion of the total heat, such a portion as experience would lead one to expect. These new diagram trials afford, in the author's (Mr. Clerk's) view, a more accurate heat-distribution balance-sheet than has yet been

obtained in any engine, from which can be deduced the ideal efficiency of the working fluid. Adding together

| | |
|---|---|
| Heat contained in gases at end of expansion | 49·3 |
| I.H.P. . . . . . . . | 34·6 |
| | 83·9 |

Then $\frac{34 \cdot 6}{83 \cdot 9} = 0 \cdot 41$. That is, if this balance-sheet be correct and the heat loss be assumed as entirely incurred at the beginning of the stroke, then the maximum efficiency of the actual working fluid for the compression and expansion is 41 per cent. of the total heat supplied." The earlier part of this quotation is the subject of our present discussion, the latter part is dealt with elsewhere.

Even with such very considerable discrepancies in the heat balance-sheets as those discussed above, the student will none the less remark that the heat utilized has now grown from about 20 per cent. to over 30 per cent. This all-important improvement has occurred therefore in spite of the many uncertainties as to how the lost heat divided itself up. It is indeed one of the fortunate features cf gas engine manufacture that improvements do not have to attend the settlement of the many intricate problems with which gas engine operation is bound up, but proceed by the trial and error of experiment with such guidance as theoretical considerations have been able to afford. The great want which has in the past caused so much theoretical difficulty has been accurate knowledge of the values of the specific heats of the working fluids.

**73. Testing of Gas Engines.**—The very scientific attitude adopted by the Germans to engineering work has led them to lay down precise rules for the testing of gas plant. These rules have been drawn up by the German Associations of Engineers, engineering firms, and large gas engine builders and published in detail in the *Zeitschrift des Vereines Deutscher Ingenieure*. Their object is to standardize procedure throughout the country and so render tests properly comparable. The more important rules are that—

1. Fuel consumption tests of gas producers are to last eight hours without interruption.
2. Fuel consumption tests on engines are to last one hour at high loads and a lesser time at low loads.
3. To determine that the engine has reached a steady state the temperature of the cooling water will be measured from time to time.
4. Mechanical efficiency tests must be taken at constant load and at least ten series of diagrams taken.
5. Temperatures to be Centigrade. One h.p.-hour to be taken as 632 calories.

**74. Engine Tests.**—The result of a test on a **200 h.p. engine and suction plant** has been published by Mr. Mathot *

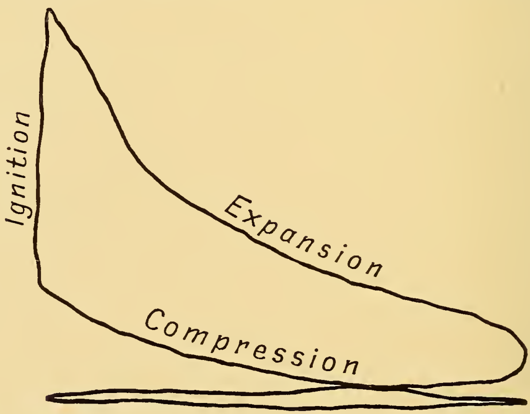



FIG. 50.—Typical Indicator Card from a Four Cycle Engine. Compare with Fig. 46.

and the more important figures are here reproduced. The engine was of the four-cycle double-acting type and was tested at the works of the well-known firm of Gasmotoren Fabrik, Deutz-Cologne.

| | |
|---|---|
| Piston diam. . . . . . . | $21\frac{1}{4}$ in. |
| ,, stroke . . . . . . | $27\frac{9}{16}$ in. |
| ,, rods diam.— | |
| Front . . . . . . . | $4\frac{3}{4}$ in. |
| Rear . . . . . . . | $4\frac{5}{16}$ in. |

* I.M.E., 1905.

FULL LOAD TESTS.

| | 1904. | |
|---|---|---|
| | March 14. | March 15. |
| Average r.p.m. | 151·29 | 150·20 |
| B.H P | 214·22 | 222·83 |
| Duration of test, hours | 3 | 10 |
| Average temperature of water after cooling piston | 117·5° F. | — |
| Average temperature of water after cooling cylinder and valve seats | 135° F. | — |
| Water consumption for cooling piston, gallons/hour | 39 | — |
| Water consumption per hour in vaporizer (anthracite fuel), galls./hour | — | 14·2 |
| Water consumption per hour in scrubbers, galls./hour | — | 318 |
| Average temperature of gas at outlet of generator | — | 558° F. |
| Average temperature of gas at outlet of scrubbers | — | 62·5° F. |
| Gross fuel consumption per b.h.p.hour | 0·727 lb.* | 0·720 lb. |
| Corresponding thermal efficiency | 19 per cent. | 24·4 per cent. |

Other interesting figures are—

Water consumption in galls. per b.h.p.-hour—

1. For cooling cylinder, stuffing boxes, valve seats and jackets . . . . . . 4·65
2. For cooling piston and piston rods . . 1·75
3. For vaporizer . . . . . . 0·0655
4. For washing the gas in the scrubbers . 1·42

Also :—

Water converted into steam per lb. of fuel consumed in generator . . . . 0·193 galls. or 1·93 lb.

In an important test carried out by Mr. J. T. Nicolson on a Crossley gas engine and suction producer plant, the calorific value of the gas was 156·5 B.T.U. as determined by analysis and 149 B.T.U. per cubic foot by Junker's calorimeter at the temperature and pressure of the calorimeter. The following measurements were made :—

* Includes fourteen hours of fires banked up.

B.H.P. = 559.

Gas per hour = 29,037 cu. ft. corrected to 0° C. and 760 mm.

Gas per b.h.p. = 51·94 cu. ft.

Heat supplied = 51·94 × 156·5 = 8,128 B.T.U. per b.h.p.-hour and thermal efficiency $= \frac{1,980,000}{778} \times \frac{1}{8,128} = \frac{2,546}{8,128} =$ 31·9 per cent.

Variation in engine speed when horse-power was instantaneously dropped from 600 to 50 was from 119·4 to 121·4 r.p.m., corresponding to a total variation of $1\frac{2}{3}$ per cent. of mean speed. No back-firing was observed to take place when this was done. These tests show remarkably good thermal efficiency and satisfactory closeness of governing. Engines of this size have not often been run on suction gas.

A third trial is that of a **150 b.h.p. six cylinder vertical gas engine** which was run for six hours on full load. The gas was taken from a pressure producer and had its calorific value measured every hour by a Simmance-Abady Calorimeter. Readings were taken every half-hour of the b.h.p.

| | |
|---|---|
| Average air temperature . . | 72·2° F. |
| Average air pressure . . | 29·56″ Hg. |
| Cu. ft. gas used per hour . | 13,000 |
| Average Calorific value (lower) . | 128·1 B.T.U. per cu. ft. |
| Engine speed . . . | 325 r.p.m. |
| B.H.P. = . . . . | 151·3. |

B.T.U. consumed by engine per b.h.p.-hour = 10,590, showing a thermal efficiency of $\frac{1,980,000}{10,590 \times 778} = 24·1$ per cent., which was up to the standard of the intended design.

**75. The Governing of Gas Engines.**—The most frequent method of governing in this country has been by means of the "**hit-and-miss**" gear, which consists of an arrangement whereby a small piece of metal normally interposed between the valve operating level and the valve spindle is moved away by the governor with the result that although the valve level continues to rock it is unable to communicate its motion to the valve. This is a very simple arrangement,

but on the Continent it is considered to be hardly sensitive enough to slight changes of speed since the valve either opens to its full amount or else does not open at all. To understand the effect produced it is only necessary to take the case of a steam engine in which the governor either closed the throttle valve altogether or else did not alter it at all. For very great uniformity of speed it is necessary to employ some governing mechanism which shall work in a more gradual manner. There is a further objection which lies against the "hit-and-miss" system in that to produce a reasonable measure of uniformity of angular velocity in the crank shaft a very heavy flywheel becomes necessary. This adds to the cost of the engine and diminishes its mechanical efficiency.

It may in fact be said that the two merits which have enabled the "hit-and-miss" gear to be used as much as it is, are its great mechanical simplicity and its ability to keep constant the proportions of gas and air in the incoming charge, so enabling the engine always to be run on its most economical mixture.

Continental makers were the first to break away from this system of governing, by arranging that the governor should produce a variable lift of the gas valve by means of a conical cam. As, however, the air supply was not interfered with this meant a continually changing richness of charge and hence a lowering of thermal efficiency. This lowering of efficiency would be attributable to the fact that there is for a given engine only one mixture which will give the best efficiency of explosion, and that as the richness changes so the time taken for the mixture to ignite will change, and therefore with a fixed ignition point the maximum pressure will not come at the most advantageous part of the stroke. The tendency now is towards a regulation of both the gas and the air supplies by throttling them after mixture, with the consequence that less weight of explosive mixture is taken in and therefore a less compression pressure is reached. Using this process it is necessary to make the compression with full mixture fairly high so that when on lighter load there shall still be enough com-

pression left to enable a sufficiently good thermal efficiency to be obtained.

**76. Flywheel Effect.**—The kinetic energy stored up in a flywheel is calculated from the following formula or one derived from it.

$$KE. = \tfrac{1}{2} I.\omega^2 \text{ ft.-lb.}$$

where $I$ = moment of inertia about the axis of revolution
and $\omega$ = angular velocity in radians per second.

From this it follows that

$$\frac{d}{d\omega}(KE.) = I.\omega.$$

For a small variation in $\omega$ compared with $KE.$ it is therefore necessary that either $I$ or $\omega$ should be large. For the ordinary purposes of industry it is sufficient to ensure that the angular velocity never varies by more than $\frac{1}{25}$th to $\frac{1}{30}$th part above or below the mean speed. For the driving of continuous current generators only half the above variations are permissible, whilst for the driving of alternators in parallel the requirements are far more stringent, involving a permissible variation of but $\frac{1}{150}$th or even in some cases $\frac{1}{200}$th part above or below the mean. Mr. Mathot,* the well-known gas engine engineer, has suggested the following formula for use in calculating the dimensions which should be given to flywheels of different types of engines :—

$$P = k\frac{N}{D^2.a.n^3}$$

where

$P$ = weight of rim (without arms or boss), in tons.
$D$ = diameter to centre of gravity of rim, in inches.
$a$ = degree of cyclic irregularity permissible.
$n$ = revolutions per minute.
$N$ = b.h.p.
$k$ = coefficient determined as below.

For Otto cycle engines, single cylinder, single acting . . . . . . . . $k = 44{,}000$

For Otto cycle engines, two opposite cylinders, single acting, or one cylinder double acting . $k = 28{,}000$

* I.M.E., 1905.

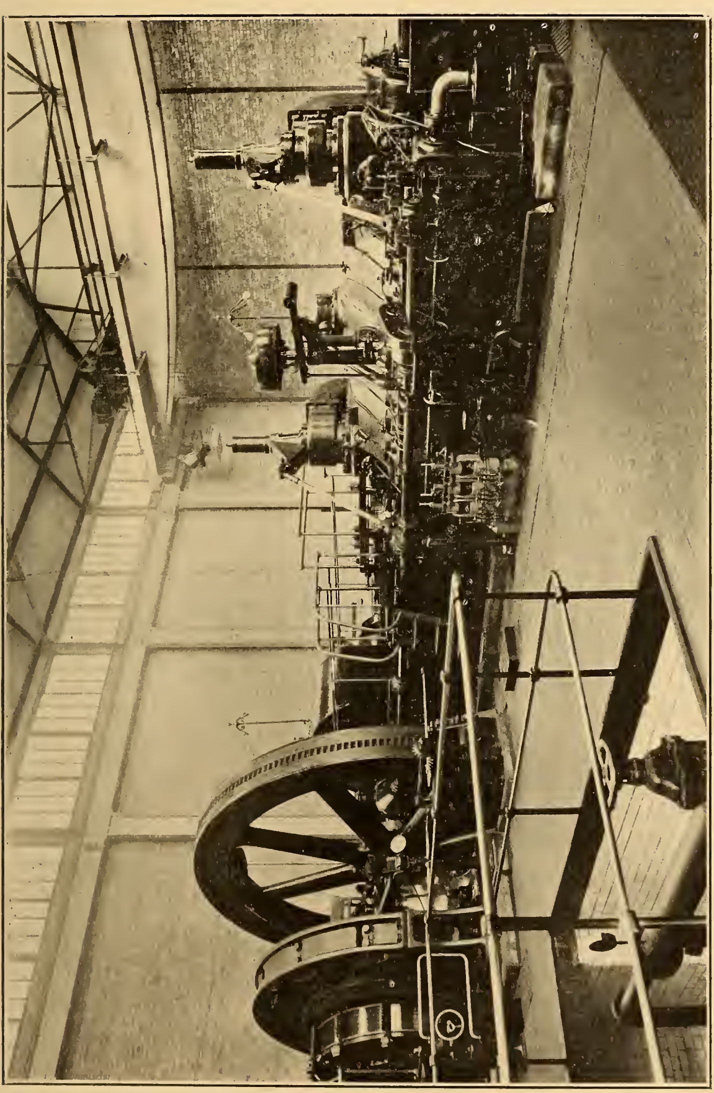


Fig. 51.—Koerting Gas Engine (Mather and Platt) driving Electric Generator.

For two cylinders, single acting, with cranks set at 90° . . . . . . $k=25{,}000$
For two twin cylinders, single acting . . $k=21{,}000$
For four twin cylinders opposite, or for two tandem cylinders, double acting . . . $k=7{,}000$

Note.—Total weight of flywheel may be put as $1 \cdot 4\, P$.

**77.** The most usual way to speak of **cyclic irregularity** is that above described. It amounts to defining cyclic irregularity as the fraction by which the instantaneous angular velocity exceeds or falls below its mean value in any one complete cycle. There is, however, another way of considering the matter. Thus Mr. L. Schüler in a paper dealing with the driving by gas engines of alternators operated in parallel remarks : " The speed of the machines should be as uniform as possible and should in any case be such that the amplitude of the angular oscillation does not exceed two electrical degrees." An electric degree means of course $\frac{1}{360}$th part of the angular distance, the passage of which by the armature corresponds to one electrical cycle. It becomes therefore a matter of interest to see how the one form of computation can be turned into the other. A good deal depends naturally on the rate at which the speed variation rises and falls, but for a sufficiently close approximation it may be taken as a sine or cosine curve. Then the angular velocity $\omega$ may be written as equal to

$$a+b \cos c\theta.$$

Where $\theta$ is the angle the crank has moved through, $a$ is the mean value of the angular velocity, and $b$ is its maximum variation from the mean, so that the speed oscillates between $(a+b)$ and $(a—b)$. If $c$ be unity this oscillation occurs once in a revolution, but if $c=2$ then it occurs twice, and so on.

This may be written

$$\omega=\frac{d\theta}{dt}=a+b \cos c\theta$$

or

$$dt=\frac{d\theta}{a+b \cos c\theta}$$

integrate and

$$c.t=A+\frac{2}{\sqrt{a^2-b^2}}\tan^{-1}\frac{\sqrt{a-b}\tan\frac{c\theta}{2}}{\sqrt{a+b}}$$

where $A$ is some constant.

Put $\theta=0$ when $t=0$ and therefore $A=0$

so that $$t=\frac{2}{c\sqrt{a^2-b^2}}\tan^{-1}\frac{\sqrt{a-b}\tan\frac{c\theta}{2}}{\sqrt{a+b}} \quad \dots \quad \dots \quad (1)$$

Now $\frac{b}{a}$ is the cyclic irregularity and may be given a name—call it $m$.

therefore $$t=\frac{2}{ac\sqrt{1-m^2}}\tan^{-1}\frac{\sqrt{1-m}\tan\frac{c\theta}{2}}{\sqrt{1+m}}$$

$$\frac{\sqrt{1-m}\tan\frac{c\theta}{2}}{\sqrt{1+m}}=\tan\frac{act\sqrt{1-m^2}}{2}$$

or $$\theta=\frac{2}{c}\tan^{-1}\left\{\frac{\sqrt{1+m}}{\sqrt{1-m}}\tan\frac{act\sqrt{1-m^2}}{2}\right\} \quad \dots \quad (2)$$

Now as $m$ is always small compared to unity $\sqrt{1+m}$ may be written as $(1+\frac{1}{2}m)$ and $m^2$ be neglected;

therefore $$\theta=\frac{2}{c}\tan^{-1}\left\{(1+m)\tan\frac{act}{2}\right\}.$$

Now were $m$ really zero this equation would give to $\theta$ a value equal to

$$=\frac{2}{c}\tan^{-1}\tan\frac{act}{2}=at.$$

Call this value $\theta_0$. Really it means the position at which the crank would be, were the angular velocity strictly uniform.

We may therefore write $\theta=\frac{2}{c}\tan^{-1}\left\{(1+m)\tan\frac{c\theta_0}{2}\right\}$ (3)

**which is the solution.**

If, for example, $c=1$ and $m=\frac{1}{200}$

then $$\theta = 2 \tan^{-1}\left\{1{\cdot}005 \tan\frac{\theta_0}{2}\right\}.$$

From this we see that when $\theta_0 = 120°$, $\theta$ becomes

$$\begin{aligned} &2 \tan^{-1}\{1{\cdot}005 \tan 60°\} \\ &= 2 \tan^{-1}(1{\cdot}005 \times 1{\cdot}73205) \\ &= 2 \tan^{-1} 1{\cdot}74071 \\ &= 2 \times 60{\cdot}12° \\ &= 120{\cdot}24° \end{aligned}$$

or that the crank would be nearly $\frac{1}{4}$ degree ahead of its true position.

It is useful to get an expression for this deviation directly. From (3)

$$\theta - \theta_0 = \frac{2}{c} \tan^{-1}\left\{(1+m)\tan\frac{c\theta_0}{2}\right\} - \theta_0.$$

Find the maximum value of $\theta - \theta_0$ by differentiating and equating to zero. Then

$$(1+m) = \left\{1 + (1+m)^2 \tan^2\frac{c\theta_0}{2}\right\}\cos^2\frac{c\theta_0}{2}$$

$$= \cos^2\frac{c\theta_0}{2} + (1+m)^2 . \sin^2\frac{c\theta_0}{2}$$

$$1 = 2\sin^2\frac{c\theta_0}{2} + m\sin^2\frac{c\theta_0}{2}$$

$$\sin^2\frac{c\theta_0}{2} = \frac{1}{2+m}$$

$$\tan^2\frac{c\theta_0}{2} = \frac{1}{1+m}$$

or $$\tan\frac{c\theta_0}{2} = 1 - \tfrac{1}{2}m \text{ approximately,}$$

so that the maximum value of $\theta - \theta_0$ is found from the expressions

$$\theta - \theta_0 = \frac{2}{c}\tan^{-1}(1 - \tfrac{1}{2}m)(1+m) - \theta_0$$

and

$$\theta_0 = \frac{2}{c}\tan^{-1}(1 - \tfrac{1}{2}m).$$

Therefore the maximum deviation =

$$=\frac{2}{c}\left\{\tan^{-1}(1+\tfrac{1}{2}m)-\tan^{-1}(1-\tfrac{1}{2}m)\right\}$$

$$=\frac{2}{c}\tan^{-1}\frac{4m}{8-m^2}$$

If $m$ be so small that $m^2$ can be neglected—as it practically is—this reduces to

$$\textbf{maximum deviation} = \frac{2}{c}\tan^{-1}\frac{m}{2} \quad \ldots \quad \ldots \quad (4)$$

If for example $c=1$ and $m=\frac{1}{25}$, this equation gives the maximum deviation

$$=2\tan^{-1}\frac{1}{50}=2\times 1{\cdot}2=2{\cdot}4 \text{ degrees.}$$

When $m$ is much smaller than $\frac{1}{25}$, say $\frac{1}{200}$ equation (4) can be approximately written :—

$\frac{2}{c}\times\frac{m}{2}=\frac{m}{c}$. So that with $c=1$ and $m=\frac{1}{200}$ the maximum deviation would be $\frac{1}{200}$ radian or 0·28 degree.

**78.** Equation (2) shows how the value of $\theta$ can be calculated for any position of the ideal crank, and the deviation may have its most important effect electrically even when it has not itself its largest numerical value. For that reason it is desirable to have some means of calculating it easily. In cases in which it is only desired to find the maximum deviation to some approximate degree of accuracy it is sufficient to take a mean value of the excess angular velocity and multiply it by the time during which it operates. Thus if as before $c=1$ and $m=\frac{1}{200}$, calling the angular velocity $\omega$, the average excess of angular velocity

$$=\frac{2}{\pi}\times\frac{\omega}{200}=\frac{\omega}{100\pi}$$

and this operates through 180° or for a time equal to $\frac{\pi}{\omega}$,

so that the angular motion gained

$$= \frac{\omega}{100\pi} \times \frac{\pi}{\omega} = \frac{1}{100}$$

and $\frac{1}{100}$ radian $= 0{\cdot}57$ deg., which, divided between the two ends of the period, gives a maximum deviation of 0·28 deg., which agrees with the 0·28 deg. previously found; but for larger values of $m$, this method would be less accurate.

For these values of $c$ and $m$ it may be said that the maximum deviation is about $\frac{1}{4}$ of a degree. If the alternator has six pairs of poles giving six electrical cycles during one mechanical one this deviation could also be called $\frac{1}{4} \times 6$ or one and a half electrical degrees, which corresponds very well with Mr. Schüler's result.

**79. Balancing.**—The problem of balancing the parts of a gas engine and providing for uniformity of torque as far as possible does not differ in principle from the corresponding problem in the case of the steam engine, and the author does not propose therefore to devote a great deal of space to this subject. The student should refer to what has been written on the subject of balancing by Professor Perry and Professor Dalby, both of whom have made a special study of the matter. It will, however, be advisable to give here a brief account of the general principles involved, leaving the application to be made to each and every problem as it presents itself. For it must be remembered that although the problem is often surrounded by complications which lead to the mathematical work looking difficult and involved, there is really no special difficulty about it at all, but merely a necessity that the fundamental principles should be rightly applied and that the algebraic or arithmetical work should be carried through without mistakes.

The simplest kind of balancing is that in which a flywheel is light on one side and requires a weight ($W$) fastened to the other side in order to prevent any jumping or vibration when the wheel rotates. This does not of necessity mean that an equal weight must be added to the other side, because it does not follow that it will be possible to place

the balance weight at the same distance from the centre of the shaft, and the centrifugal force being equal to $\omega^2.r.\frac{W}{g}$ (where $\omega$=angular velocity in radians per second and $r$= distance in feet from the centre of the shaft) it is evident that the product of the $W$ and the $r$ in the balance weight must come out to a certain amount. If therefore the $r$ is very small then $W$ must be proportionately greater, and inasmuch as the balance weight is often bolted in between the spokes it is clear that $r$ will usually be less than the radius of the rim of the wheel. This is the simplest kind of balancing. The most complicated kind occurs in the motion of a rod, like a connecting-rod, in which one end reciprocates to and fro in a straight line and the other end follows a circular path, with the result that intermediate parts of the rod follow a complicated curve and one not easy to treat. In such a case as this it is customary to obtain an approximate solution by assuming that a certain part of the rod is massed at the crosshead and the rest at the crank-pin, and it is not unusual to make this division of the rod in inverse proportion to the distance of the centre of gravity from either end. This is only an approximation unless it happens that the rod is so made (which it usually is not) that if hung up from the big and little ends in turn it will swing, pendulum wise, with the same number of swing-swangs per minute. When a number of rotating masses (real or assumed) have to be balanced it is useful, following Dalby's method, to consider the plane perpendicular to the shaft in which one or more of them lie to be rotating at the same speed as the shaft and to draw out on this plane the force diagram.

**80.** The connecting rod comes into the problem in yet another way. If the crank-pin rotated uniformly and the connecting rod were exceedingly long the motion of the piston would be a Simple Harmonic Motion (usually written S.H.M.) and the displacement of the piston from the middle of its stroke would be equal to $r. \cos \theta$ where $r$=radius of crank-pin circle and $\theta$ is the angle between the crank and the line of dead centres. But the connecting rod in actual

engines is usually quite short, never more than ten times the length of the crank arm and usually much less. This produces a complicated motion of the piston, and it will be useful to calculate exactly what it is. Let $P$ be the crank-pin and $A$ the piston which, in the position shown, is at a distance $AB$ from the beginning of its stroke. The angles $\theta$ and $\phi$ are as shown in the diagram. $OP$ is $r$ and $AP$, $l$. $AB$ will be written as $x$. Now it is clear that

$$r \cos \theta + l \cos \phi + x = BO$$

and $$BO = l + r$$

so that $$r \cos \theta + l \cos \phi + x = l + r \quad \ldots \quad (1)$$

Also we have $$r \sin \theta = l \sin \phi \quad \ldots \quad (2)$$

It is necessary to combine these two equations so as to find $x$ in terms of known quantities.

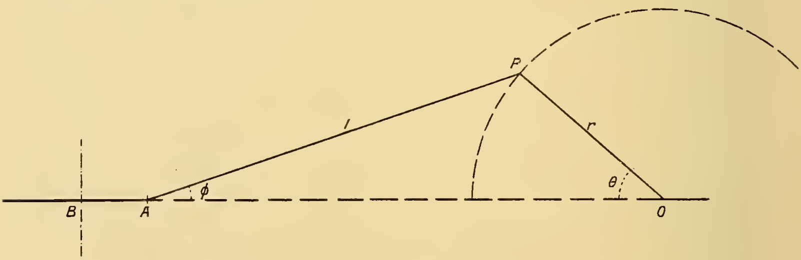



FIG. 52.—Motion of Crank-pin and Connecting Rod.

From (1)

$$\begin{aligned} x &= l + r - r \cos \theta - l \cos \phi \\ &= l(1 - \cos\phi) + r(1 - \cos \theta) \end{aligned}$$

Also from (2)

$$\sin \phi = \frac{r}{l} \sin \theta$$

and $$\cos \phi = \sqrt{1 - \frac{r^2}{l^2} \sin^2 \theta}$$

therefore $$x = l\left(1 - \sqrt{1 - \frac{r^2}{l^2} \sin^2\theta}\right) + r(1 - \cos \theta) \quad \ldots \quad (3)$$

and this gives the value of $x$ for any value of $\theta$.

Previously we have spoken of the distance of the piston from the mid point of its stroke rather than from either end, and it is useful to follow the same procedure here—

call the displacement of the piston from mid stroke $y$—then $x+y=r$

or
$$y=r-x$$

so that
$$y=r\cos\theta-l\left(1-\sqrt{1-\frac{r^2}{l^2}\sin^2\theta}\right)$$

furthermore $\theta$ is a function of the time, and as uniformity of rotation is assumed it will be directly proportional to the time. Put therefore $\theta=\omega t$

so that $y=r\cos\omega t-l\left(1-\sqrt{1-\frac{r^2}{l^2}\sin^2\omega t}\right)$ .. .. (4)

Since, however $\left(\frac{r}{l}\right)^2$ is a small amount in all engines an approximation to the above may be written as

$$y=r\cos\omega t-l\left(1-1+\tfrac{1}{2}\frac{r^2}{l^2}\sin^2\omega t\right)$$

$$=r\cos\omega t-\frac{r^2}{2l}\sin^2\omega t$$

or
$$y=r\cos\omega t-\frac{r^2}{4l}(1-\cos 2\omega t) \quad .. \quad .. \quad .. \quad (5)$$

This very interesting result shows that the position of the piston can be stated as the sum of two S.H.M.'s one of which corresponds to an infinitely long connecting rod and the other to a S.H.M. of twice the periodicity and of an amplitude depending on the ratio of $r$ to $l$. The motion in fact is analogous to that of the air set into vibration by an organ pipe which in addition to giving its fundamental note gives also a weak first harmonic. Although this first harmonic is weak in its effect on the displacement of the piston it is considerably more potent when velocities and accelerations have to be taken into account, as will presently appear.

Since from (5)

$$y=r\cos\omega t-\frac{r^2}{4l}(1-\cos 2\omega t)$$

$$\frac{dy}{dt}=-\omega r\sin\omega t-\frac{r^2\omega}{2l}\sin 2\omega t$$

$$=-\omega r\left(\sin\,\omega t+\frac{r}{2l}\sin\,2\omega t\right)\quad ..\quad ..\qquad (6)$$

and $$\frac{d^2y}{dt^2}=-\omega^2 r\left(\cos\,\omega t+\frac{r}{l}\cos\,2\omega t\right)\quad ..\quad ..\qquad (7)$$

It is important to note that expression (6) which gives the velocity of the piston at any point has the multiplier $\frac{r}{2l}$ in front of the harmonic term, and that expression (7) which gives the acceleration and therefore measures all the inertia forces has the multiplier $\frac{r}{l}$. It follows therefore that the three multipliers in the harmonic term for displacement, velocity and acceleration run thus, $\frac{r}{4l}$, $\frac{r}{2l}$ and $\frac{r}{l}$, showing that a ratio of $\frac{r}{l}$ which will produce a 5 per cent. difference in the position of the piston will bring about a 10 per cent. change in the velocity and about 20 per cent. in the acceleration. It will now be realized that, when forces are being nicely balanced, the growing importance of the harmonic term must be carefully allowed for.

It is often useful to bear in mind a simple rule for the value of the acceleration at the ends of the stroke, i.e. when $\omega t=0°$ or $180°$. From formula (7) it will be seen that this leads to

$$\frac{d^2y}{dt^2}\text{ either}=-\,\omega^2 r\left(1+\frac{r}{l}\right),\text{ or}=-\omega^2 r\left(-1+\frac{r}{l}\right)$$

$$=\mp\,\omega^2 r\left(1\pm\frac{r}{l}\right)$$

a very simple approximate rule, i.e. that the acceleration at the end of the stroke is more or less than the S.H.M. value by the fraction $\frac{r}{l}$ of that value.

**81. Connecting Rod Effect.**—This is best illustrated by a geometrical construction due to Mr. J. Harrison, M.I.C.E.

The construction is as follows :—

*OB* is the crank and *AB* the connecting rod of which *G* is the centre of gravity.

*OQ* and *SH* are perpendicular to *AO*.

*Ha* is perpendicular to *AB*.

*SQ* and *Gg* are parallel to *AO*.

$GU = k^2/AG$ where $k$=radius of gyration.

*UX* is parallel to *Ba*.

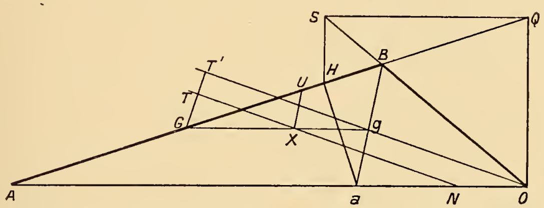



FIG. 53.—Resultant of all the Accelerating Forces on a Connecting Rod.

Then *TXN* parallel to *gO* is the line of action of the resultant of all the forces acting on the rod and its value in $mq^2.XN$ where $m$=mass of rod and $q$=angular velocity of crank-pin. The proof of this construction is given in Professor Perry's *Steam Engine*, p. 549, and may there be referred to by those interested.

This diagram enables the direction and amount of the inertia forces due to the connecting rod to be calculated for each position of the crank.

If $k^2$ happens to be equal to the product of *AG* and *GB* then $GU = k^2/AG = GB$ so that the point *U* would coincide with *B* and the resultant force would pass through *O* and hence there would be no "whipping effect" of the rod. One often sees connecting rods produced beyond the crank-pin with the object of bringing about this relationship. When it is accurately obtained it will be found that the period of swing of the rod about either big or little end will be the same.

## PROBLEMS.

1. Describe a gas engine, and how it uses the Otto cycle of operations. Sketch the cylinder, showing piston, water-

jacket, valves, shape of clearance space, and shape of exhaust outside the cylinder. We do not want sketches of any of the other parts of the engine, although they are mentioned in the description. Draw the usual sort of diagram to scale. (B. of E., 1899.)

2. Describe the cycle of operations in a common gas engine. Sketch and describe the construction and action of the mechanism by which the speed of the engine is controlled. What would be the i.h.p. of such an engine whose piston is 12 in. diameter, its crank is 8 in. long, the engine makes 160 revolutions or 80 cycles per minute, and 30 per cent. of the possible explosions are omitted? The mean area of all the diagrams on a card taken with a 120 spring in the indicator as measured by the planimeter is 2·62 sq. in.; length of diagram parallel to atmospheric line 4·03 in.

*Ans.* 19·95 h.p. (B. of E., 1899.)

3. Describe, with sketches, *either* a gas or oil engine, and show by a diagram how it uses the Otto cycle of operations. Sketch the cylinder, showing piston, water-jacket, valves, shape of clearance space, and how the exhaust is provided for. (B. of E., 1899.)

4. What is meant by "scavenging" in relation to gas engines? How is it done, and how (or why) does it affect the efficiency? (Mech. Sc. Tripos, Part I, 1898.)

5. Sketch in section a gas engine cylinder, showing the valves and piston. (B. of E., 1900.)

6. Sketch a section through the gas valve of a gas engine, showing the hit-and-miss mechanism operated by the governor. (B. of E., 1907.)

7. The mean effective pressure on the piston, both in the forward and back strokes, is 62 lb. per square inch; cylinder 18 in. diameter; crank, 18 in. long. What is the work done in one revolution? *Ans.* 94,660 ft.-lb. (B. of E., 1906.)

8. Sketch a gas engine indicator diagram.

How is it used in finding the indicated horse-power? State clearly what information is necessary.

Why must we know the number of explosions per minute rather than the number of revolutions?

(B. of E., 1900.)

9. Describe, with sketches, how lubrication of the various parts of an engine (not encased) is now usually performed. (B. of E., 1902.)

10. Assuming that an engine works against a constant resistance and that the energy of the moving parts of the engine can, at any instant, be represented by the expression $\frac{1}{2}(M+mk^2)V^2$. where $V$ is the crank-pin velocity and $k$ is the ratio between the instantaneous velocities of the piston and the crank-pin, show how the variations of crank-pin velocity during a revolution may be deduced from the crank-effort diagram.

(Mech. Sc. Tripos, Part II, 1898.)

11. A flywheel has moment of inertia $I$; two wheels, one on each side of the first, have moments of inertia $i$ each. Each length of shaft has the same stiffness $s$. One of the smaller wheels has applied to it a varying torque $a \sin 2\pi ft$, and the other $a \cos 2\pi ft$. Neglect the mass of the shaft. What value of $f$ will cause fracture of the shaft? Why is this not exact as an illustration of what occurs between two cranks of a steam or gas engine and a flywheel midway between them? (B. of E., 1907.)

12. Why do we regulate an engine with both a flywheel and a governor? Explain clearly how each effects the regulation. (B. of E., 1900.)

13. Two engines with the same centre line on opposite sides of a crank shaft; same moving masses; cranks exactly opposite, so that there is exact balance of horizontal inertia forces; what may be done to the connecting rods to make perfect inertia balance? Prove your statement. Is the engine perfectly balanced now?.

(B. of E., 1901.)

14. Crank 1 ft., connecting rod 4·5 ft.; what are the accelerations at the ends and some other point in the stroke, if the engine makes 200 revolutions per minute? The piston and rod and crosshead are 420 lb.; draw a diagram to show the force in pounds required to produce the motion. State the scale clearly. (B. of E., 1906.)

15. A piston and rod and crosshead weigh 330 lb. At a certain instant, when the resultant total force due to steam

pressure is 3 tons, the piston has an acceleration of 370 ft. per second in the same direction. What is the actual force acting at the crosshead? (B. of E., 1902.)

16. Give an account of the different methods used for governing gas engines, stating the advantages and disadvantages of each. (Mech. Sc. Tripos, Part I, 1904.)

17. What are the chief sources of loss in the plant consisting of boiler, reciprocating engine and condenser? What is the greatest proportion of the heat given to an engine that can theoretically be turned into work, and under what conditions can this maximum be reached?

Show in what respect gas engines, oil engines and steam turbines approach more nearly to these conditions than the ordinary reciprocating engine.

(Old Regulations Cambridge B.A. Degree, 1904.)

# CHAPTER VI

# The Gas Producer

Theory—Typical Suction and Pressure Producers—Tests—Costs—Use of Gas Producer for Marine Purposes—Appendix containing Description of Mode of Operation of Suction Gas Plant.

**82. Producer Gas.** *Theory.*—In a steam boiler the energy stored up in the coal is liberated by combustion in an atmosphere containing oxygen. In other words, heat is liberated by the combination of the carbon with oxygen first to form CO, and then, if enough air be present to add a further atom of oxygen to the molecule, to $CO_2$. When 12 kg. of carbon (that is to say the atomic weight of carbon taken in kilograms) are oxidized to CO, 29,400 calories are given off, and when $CO_2$ is formed a further 68,200 calories are liberated, making a total of 97,600. This means that if the carbon be only oxidized to the CO stage not more than about 30 per cent. of the available heat energy is given up, and that by far the most of the available heat is obtained from the stage in which CO becomes $CO_2$. Even supposing that in a given steam boiler the whole of the 97,600 calories were given off from each 12 kg. of carbon (neglecting for the moment the hydrogen and hydrocarbons in the coal) only a fraction, not greater than 60 per cent., ever gets to the water, and the balance goes away up the chimney or is lost by radiation. With gas producers, however, no such losses occur. Their efficiency depends upon the working process, but it may be taken as being seldom less than 80 per cent. and often as much as 90 per cent. even when working with **anthracite coal** and not chemically pure carbon. In a gas producer, air is forced or drawn through a mass of highly heated fuel, with the result that the carbon is oxidized.

Also, in order to keep the temperature within reasonable limits, and for another reason to be given later, steam is admitted along with the air and both together pass upwards through the glowing fuel.

When the air and steam are forced through by pressure the producer is called a **Pressure Producer.** When however they are drawn through by suction caused by the suction strokes of the engine, they are known as **Suction Producers.** The theory is in each case the same.

It may seem strange that the gas given off should contain as much as 80 to 90 per cent. of the total heat energy in the coal. Those who approach the subject for the first time are aware from their knowledge of chemistry that even if pure CO came away from the producer, and no $CO_2$ at all, there would even then be a loss of the 30 per cent. of energy given up when the carbon was oxidized to CO, so leading to an **apparent maximum possible efficiency of 70 per cent.** The explanation is that the 30 per cent. is not wasted. It serves to keep the furnace alight, and to decompose the entering steam. This steam is decomposed into hydrogen and oxygen thus

$$2H_2O = 2H_2 + O_2 \quad \ldots \quad \ldots \quad \ldots \quad (1)$$

and in so doing it stores up 116,400 calories for each 36 kg. of water decomposed. It is not difficult to see that by balancing the proportions of air and steam admitted it is possible to absorb the whole of the 30 per cent. of energy rendered available by the formation of CO, and to carry it as potential chemical energy to the gas engine where the hydrogen and oxygen can again unite. In reality it is not quite so simple as this, because the oxygen from the decomposed steam has also to pass over glowing carbon, with the result that a further supply of CO is formed. Radiation of heat occurs also, and this prevents the efficiency being 100 per cent.

Following generally the procedure adopted by Mr. Dowson, who invented the first of these plants, the reactions may be semi-mathematically stated thus :—

Taking weights equal to molecular weights in kg.

Carbon-monoxide is thus formed :—

$$2C+O_2=2CO+58,800 \text{ Calories} \quad .. \quad .. \quad (2)$$

Carbon-dioxide would be formed thus :—

$$C+O_2=CO_2+97,600 \text{ Calories} \quad .. \quad .. \quad (3)$$

The former of these two equations gives a gas having a calorific value of about 119 B.T.U. per cubic foot, but when **steam** is admitted this value rises rapidly owing to the hydrogen present.

As already stated the decomposition of steam follows

$$2H_2O=2H_2+O_2—116,400 \text{ calories} \quad .. \quad (4)$$

the negative sign meaning that heat is absorbed and not liberated. The oxygen so produced also joins in the reaction, so that one of the following formulae

$$H_2O+C=H_2+CO—28,800 \text{ calories} \quad .. \quad (5)$$

$$\text{or } 2H_2O+C=2H_2+CO_2—18,800 \text{ calories} \quad .. \quad (6)$$

is followed in the decomposition of the steam. In both (5) and (6) an absorption of heat takes place which allows of a balance being obtained by a careful regulation of the relative proportions of air and steam admitted.

**83.** It is useful to discover **what quantity of water is theoretically required** per pound of coal in order to keep this reaction balanced.

Assume that the reaction follows equations (2) and (6). Really it will not follow quite such simple laws, but it will approximate thereto if the temperature is high enough. Equation (2) shows that for each 24 kg. of carbon used 58,800 calories will be liberated, and equation (6) that 18,800 calories will be absorbed by each 36 kg. of steam dissociated, requiring also for its dissociation 12 kg. of carbon. To absorb the whole of the 58,800 calories liberated $36 \times \frac{58,800}{18,800}$ kg. of steam would be required. But the steam is not admitted to the producer as steam, but as water, and there is therefore the latent heat of evaporation to be considered. Now the latent heat of 36 kg. of water-vapour at 20° C. is 21,600 calories, and this must be added to the 18,800 calories due to chemical dissociation, making a total of 40,400 calories, so that only

$36 \times \frac{58,800}{40,400}$ kg. of water would really be required, and this works out at 52·4 kg. of water. The quantity of carbon corresponding to this is clearly $24 + \left(\frac{12}{36} \times 52{\cdot}4\right) = 24 + 17{\cdot}5 = 41{\cdot}5$ kg. of carbon. So that $\frac{52{\cdot}4}{41{\cdot}5}$ or 1·26 kg. of water *will be required for each kg. of carbon.*

The next point to determine is the nature of the mixture of gases given off in this way. Equation (2) shows that for each 24 kg. of carbon there will be given * off $22{\cdot}4 \times 2 \times 1,000$ litres $= 44,800$ litres of CO. Equation (6) adds to this an equal volume of hydrogen and half the volume of $CO_2$ for each 12 kg. of carbon. Now the quantities in equation (6) must clearly be proportional to 17·5 and not 12 kg. of carbon, and therefore the volume of hydrogen will be $\frac{17{\cdot}5}{12} \times 44,800 = 65,200$ litres and the volume of $CO_2$ will be 32,600 litres. The total will therefore be

| | | |
|---|---|---|
| CO :— | 44,800 | litres |
| $CO_2$ :— | 32,600 | ,, |
| $H_2$ :— | 65,200 | ,, |
| | 142,600 | litres or 142·6 cubic metres. |

But it must be remembered that in equation (2) oxygen is supplied to the extent of 22,400 litres, and that as this is drawn from the air it will be accompanied by $\frac{79}{21} \times 22,400$ litres of nitrogen which will pass through without change. So that to the above table must be added $\frac{79}{21} \times 22,400 = 84,100$ litres of nitrogen, making the total and proportions thus :—

* Based on the principle that the molecular weight of any gas taken in grams will occupy a volume of 22·4 litres. (Some recent work has been based on a revised figure of 22.25 litres.)

| | | | |
|---|---|---|---|
| $CO$ . . | 44,800 | litres | or 19·8 per cent. |
| $CO_2$ . . | 32,600 | ,, | or 14·4 per cent. |
| $H_2$ . . | 65,200 | ,, | or 28·8 per cent. |
| $N_2$ . . | 84,100 | ,, | or 37·0 per cent. |
| | 226,700 | | 100·0 |

Thus 226,700 litres of gas are given off for each 41·5 kg. of carbon or 5,450 litres per kg. of carbon, and 5,450 litres is of course 5·45 cubic metres.

What is the **calorific power of this producer gas**? The $N_2$ and $CO_2$ can do nothing. The CO will yield up (97,600—29,400) calories for each 28 kg. of CO, or $\frac{68,200}{28}$ $=2,440$ calories per kg. of CO. The $H_2$ will yield 116,400 calories per 4 kg. of gas, or 29,100 calories per kg. of hydrogen. Take 1 cubic metre or 1,000 litres of the producer gas. It will contain 198 litres of CO yielding $\frac{198}{22,400}\times 68,200$ $=602$ calories, and of hydrogen $\frac{288}{22,400}\times 58,200=750$ calories, making a total of 1,352 calories per cubic metre. Furthermore the steam formed by the union of the hydrogen and oxygen will be capable of yielding up its latent heat which will add 21,600 calories for each 4 kg. of hydrogen concerned. Now the weight of the hydrogen in 1,000 litres of the gas is $\frac{288}{22,400}\times 2$ kg. and the calories in the latent heat of the steam will therefore be $\frac{288}{22,400}\times 2\times\frac{21,600}{4}=139$ calories, which when added to the 1,352 calories found above, makes a total calorific value of 1,491 calories per cubic metre of the gas given off by the producer. In cases in which the latent heat of the steam formed cannot be utilized, it is customary to use the lesser value of the calorific constant, and write it down in this case as 1,352 calories only, which is nearly 10 per cent. less. The figure of 1,491 calories per cubic metre corresponds to **168 B.T.U. per cubic foot.**

**84.** Dowson has carried out calculations similar to the

above for a number of possible reactions, and the following tables show some of the results he has found.

| Reaction between Air and Carbon: proportions of CO and $CO_2$ formed per cent. by volume, depending upon the temperature of the reaction. | | Composition of gas per cent. by volume. (Steam decomposed according to equation (6)). | | | | Steam used per kilo of Carbon | Gas formed per kilo of Carbon | Calorific Power of Gas made. | |
|---|---|---|---|---|---|---|---|---|---|
| CO. | $CO_2$. | $CO_2$. | CO. | $H_2$. | $N_2$. | Kilos. | Cubic Metres. | Calories per Cubic Metre. | B.T.U. per Cubic Foot. |
| 0 | 100 | 28·45 | — | 40·25 | 31·3 | 2·12 | 6·54 | 1,243 | 139·7 |
| 10 | 90 | 27·8 | 0·9 | 39·7 | 31·6 | 2·08 | 6·48 | 1,254 | 140·9 |
| 20 | 80 | 27·1 | 1·9 | 39·15 | 36·85 | 2·02 | 6·41 | 1,267 | 142·4 |
| 30 | 70 | 26·3 | 3·0 | 38·5 | 32·2 | 1·97 | 6·34 | 1,282 | 144·0 |
| 40 | 60 | 25·35 | 4·3 | 37·7 | 32·65 | 1·19 | 6·26 | 1,298 | 145·8 |
| 50 | 50 | 24·3 | 5·85 | 36·8 | 33·05 | 1·83 | 6·17 | 1,316 | 147·9 |
| 60 | 40 | 23·0 | 7·65 | 35·8 | 33·55 | 1·75 | 6·07 | 1,340 | 150·5 |
| 70 | 30 | 21·5 | 9·8 | 34·55 | 34·15 | 1·66 | 5·95 | 1,366 | 153·5 |
| 80 | 20 | 19·6 | 12·4 | 33·0 | 35·0 | 1·55 | 5·81 | 1,398 | 157·1 |
| 90 | 10 | 17·3 | 15·65 | 31·1 | 35·95 | 1·42 | 5·65 | 1,438 | 161·6 |
| 100 | 0 | 14·4 | 19·7 | 28·8 | 37·1 | 1·26 | 5·45 | 1,490 | 167·5 |

This table serves to show the very thorough manner in which Mr. Dowson has worked out the chemical problems relating to producer gas, and the student who wishes to pursue such matters further is referred to that writer's very able book on the subject.

We have now discussed the ideal conditions of working. In practice about the theoretical weight of water is used in suction producers. For pressure producers such as the Mond producers an excess of steam is admitted in order that the temperature of the coal may be kept to a point lower than that at which **ammonia** dissociates, it being a feature of this process to recover and sell the ammonia produced from the nitrogen contained in **bituminous coals**; the

effect of this, incidentally, is to lower the thermal efficiency of the producer to about 80 per cent.

Equation (5) may sometimes be followed instead of equation (6) for the decomposition of the steam, depending on the temperature of the reaction and the masses involved. Mr. Dowson gives these two comparisons of the theory and practice in each case :—

**Theory.**

Gas formed according to Equations (2) and (5) :—

| | Per cent. by Volumes. |
|---|---|
| $CO$ | 39·9 |
| $H_2$ | 17·0 |
| $N_2$ | 43·1 |
| | 100·0 |

**Practice.**

Gas made at Millwall. 121·3 vols. contain same weight of carbon and consist of :—

| | Volumes. |
|---|---|
| $CO$ | 33·5 |
| $H_2$ | 18·6 |
| $N_2$ | 62·8 |
| $CO_2$ | 4·7 |
| Methane | 1·7 |
| | 121·3 |

Gas formed according to Equations (2) and (6) :—

| | Per cent. by Volumes. |
|---|---|
| $CO$ | 19·7 |
| $H_2$ | 28·8 |
| $CO_2$ | 14·4 |
| $N_2$ | 37·1 |
| | 100·0 |

It will be noticed that an excess of air has been admitted in each case.

Gas made at Winnington. 117·6 vols. contain same weight of carbon and consist of :—

| | Volumes. |
|---|---|
| $CO$ | 12·9 |
| $H_2$ | 34·1 |
| $CO_2$ | 18·8 |
| $N_2$ | 49·4 |
| Methane | 2·4 |
| | 117·6 |

**85. Actual Producers.** In Fig. 54 is shown a reproduction of a working drawing of a 150 h.p. suction producer made by the Campbell Gas Engine Co. The steam required for the reaction is derived from the annular boiler surrounding the gas producer, and the heat necessary for vaporization is derived from the heat of the fuel. This steam passes with the air down a pipe leading to the base of the gas producer, and is then drawn through the glowing fuel which is maintained at a temperature of about 1,000° C. The air and steam on passing through the furnace are decomposed in accordance with the

equations already given, and the hot producer gas then passes through a dust trap or separator, and then past a

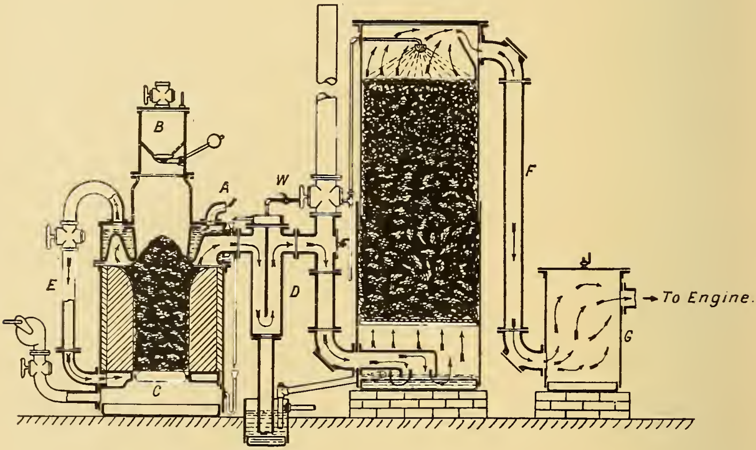



FIG. 54.—Sectional elevation of a 150 H.P. Campbell Suction Gas Producer. Fuel is first admitted through the hopper *B*. Air then enters at *A*, picks up steam on its way and passes by the pipe *E* to the grate *C*. The gases come away from the upper part of the producer and pass by the pipe system shown at *D* to the Scrubber Chamber where they are cleansed and cooled. The gases are next drawn along the pipe *F* to the expansion box *G* on their way to the engine.

water seal into the coke scrubber which consists of a tall vertical vessel containing coke upon which a water spray is kept playing. This cools the gas, condenses any steam there may be in it and serves generally to cleanse it. Thence the gas passes to a gas box to equalize the pressure, and from that it is drawn into the engine as wanted. A full description of how to work such a producer is, on account of its general interest to the many readers who will be unacquainted with the actual working of such plant, given as an appendix to this chapter. The above description applies to a plant using anthracite. When it is desired to use coke, a sawdust scrubber is usually required in addition to the coke scrubber. An outside view of a similar plant is also given in Fig. 55.

There is not a great deal of difference between the different makes of suction producer plant. Fig. 56 shows an outside view of a National Gas Engine Co. type, similar to that

which was awarded the gold medal at the Royal Agricultural Society's Trials in 1906. Its internal arrangements are very similar to those already described, except that the vaporizer is fed with water which has first been heated by being passed through a pipe in the gas outflow passage and is then vaporized on the "flash" system.

Pressure producers are worked on much the same general principles, except that the air and steam are forced through the coal instead of being sucked through. In general, too,

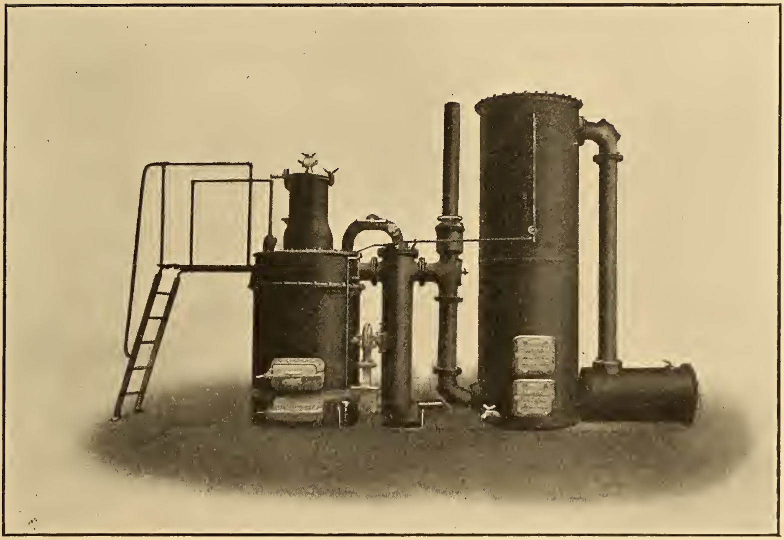


FIG. 55.—Outside view of 80 B.H.P. Campbell Suction Gas Plant. Note small size of Producer for the amount of power produced.

they are for much larger plants. Suction producers are usually for quite small outputs—commonly about 30 or 40 h.p. and rarely going beyond 500 h.p., whereas the power from pressure producers may run into thousands of horse-power and the latter are therefore of a much more extensive nature, and a good deal more complicated, especially when a feature is made of by-product recovery.

**86. Tests.** It will be of interest to give here some figures from the tests held on suction producer plant in 1905

by the Highland and Agricultural Society of Scotland, and in 1906 by the Royal Agricultural Society.

In the 1905 trials ten complete plants, exhibited by six

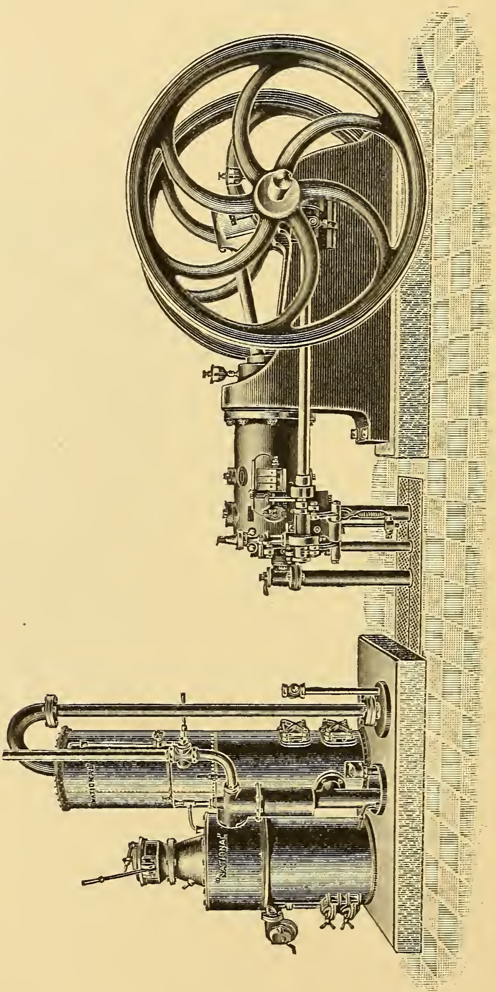


Fig. 56.—National Gas Engine and Suction Plant.

different firms, were sent in for the competition. Particulars of these plants are given in the following table—

GENERAL DIMENSIONS AND PARTICULARS OF SUCTION-GAS PLANTS ENTERED FOR TRIAL.

| — | Plants of about 20 Brake Horse-Power Capacity. | | | | | | Plants of about 8 Brake Horse-Power Capacity. | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| Exhibitor | The Acme Engine Company, Shettleston, Glasgow. | The Campbell Gas-Engine Company, Halifax. | Messrs. Crossley Bros., Openshaw, Manchester. | The Industrial Engineering Company, Hyde, near Manchester | The National Gas-Engine Company, Ashton-under-Lyne. | Messrs. Tangyes, Ltd., Birmingham. | The Campbell Gas-Engine Company, Halifax. | The Industrial Engineering Company, Hyde, near Manchester | The National Gas-Engine Company, Ashton-under-Lyne. | Messrs. Tangyes, Ltd., Birmingham. |
| Declared capacity of gas-producer plant, B.H.P. | 25 | 18 | 24 | 25 | 20 | 21 | 8 | 10 | 10 | 12 |
| Price of gas-producer plant (complete) | £94 10*s.* | £105 | £80 | £72 | £80 | £90 | £80 | £60 | £65 | £58 |
| Total weight of plant, cwt. | 35 | 25 | 45 | 25 | 25 | 28 | 25 | 15 | 20 | 17 |
| General description of engine | "Acme" | Campbell | Crossley | "Acme" | "National" | Tangye | Campbell | "Acme" | "National" | Tangye |
| Declared brake horse-power of engine | 22 | 18 | 16 | 22 | 20 | 19 | 8 | $13\frac{1}{2}$ | 8 | $7\frac{1}{2}$ |
| Diameter of cylinder, in. | 10 | $9\frac{1}{2}$ | $8\frac{1}{2}$ | 10 | 10 | 10 | 7 | $8\frac{1}{2}$ | 7 | 7 |
| Stroke ,, | 17 | 18 | 20 | 17 | 18 | 19 | 12 | 14 | 15 | 16 |
| Revolutions per minute (declared) | 220 | 200 | 200 | 220 | 190 | 190 | 230 | 230 | 220 | 220 |
| Price of engine (complete) | £130 | £125 | £110 | £130 | £120 | £131 | £80 | £75 | £80 | £90 |
| Weight of engine, cwt. | 60 | 66 | 54 | 60 | 78 | 69 | 29 | 25 | 40 | 40 |
| Price of producer plant and engine (complete) | £224 10*s.* | £230 | £190 | £202 | £200 | £221 | £160 | £135 | £145 | £148 |
| Space taken up by complete plant—producer, engine, etc. sq. ft. | 144 | 225 | 240 | 288 | 225 | 225 | 160 | 288 | 225 | 160 |

The result of the trials was given in the Judges' report,* of which the following contains an account.

Each plant was allowed half an hour of steady working before the actual power test, at the end of which the plant was brought back as nearly as possible to the same condition in respect of fuel, etc., as it was at the beginning of the trial, and the actual weight of fuel supplied in the interval was taken as that consumed by the plant during the power test. The obviously weak point in this procedure was that it was quite impossible to determine absolutely whether the plant was really in the same condition at the end of the trial as it was at the beginning. By running the test for a long enough time, however, any slight error in this respect could be rendered of little importance, and probably the method adopted was the best one. The alternative would have been to start the producers up from rest, and note the fuel put in, then at the end of the trial, note the proportion in the producer which had not been burnt, subtract the two, and add to this any fuel which had been introduced during the test. This procedure was adopted at the R.A.S. trials in 1906, except that the fuel consumed when the producers were banked up all night was also included, so leading to the disadvantage that it did not give a real fuel economy test. Also it was extremely difficult to tell at the end of the trial how much of the fuel left in the producer could properly be said to be "unburnt."

In the Scotch trials it was found that the coal used per brake h.p. at full load, varied from 1·25 to 0·84 lb., and at half load from 1·55 to 0·91 lb. This was for the 8 h.p. sizes. For the larger, 20 h.p., plants the fuel used per b.h.p. at full load varied from 0·93 to 0·77 lb. and at half load from 1·08 to 0·92 lb. These results serve to show how economical the suction producer plant is when compared with steam engine plant of the same output; the latter would consume anything from 2½ times to 4 times as much fuel per b.h.p. Other interesting figures reported by the Judges are that the capacity of the producer per declared b.h.p. varied from 0·124 cu. ft. to 0·295 cu. ft. for the 20 h.p. size, and from 0·161

* *Engineering*, November 17, 1905.

cu. ft. to 0·372 cu. ft. for the 8 h.p. size. Each of these figures show a ratio of about 2·3 to 1 and the price of the plants varied also but in not so great a ratio. The variation in cubic feet capacity per b.h.p. was sufficient indication that little had been done towards standardization of design.

**87.** The tests carried out by the R.A.S. in 1906 were considerably more elaborate, and, as already stated, a different procedure was followed. The report of the Judges had been published and, although in some aspects it may be said to be controversial, it is certainly worth study. Fourteen plants were entered for trial and all but three ran through to the finish. The capacity in each case was 15 to 20 h.p. A list of the plants with their leading dimensions and other particulars is given here—

| Name of Producer. | Name of Engine. | Revs./min. | Stroke In. | Diam. of Cyl. In. | Declared b.h.p. on Anthracite |
|---|---|---|---|---|---|
| National . . . | National . . | 190 | 18 | 10 | 20 |
| Dowson . . . | Railway and General | 170 | 18 | 12 | 20 |
| Paxman . . . | Paxman | 220 | 15 | 9½ | 15·5 |
| Dowson . . | National . . | 190 | 18 | 10 | 20 |
| Campbell. . . | Campbell . . | 200 | 19 | 9½ | 18 |
| Campbell. . . | Campbell . . | 190 | 20 | 10 | 20 |
| Dudbridge . . | Dudbridge . . | 200 | 17 | 9¾ | 20 |
| Mersey . . . | Gardner . . | 200 | 18 | 9 | 20 |
| Hindley . . . | Hindley. . . | 600 | 7 | 7 | 16 |
| Kynoch . . . | Kynoch. . . | 240 | 18 | 9 | 17 |
| Newton . . . | Newton. . . | 200 | 18 | 9 | 20 |
| Fielding . . . | Fielding. . . | 220 | 18 | 9½ | 18 |
| Crossley . . . | Crossley. . . | 220 | 21 | 8½ | 17 |
| Crossley . . . | Crossley. . . | 180 | 21 | 8½ | 15 |

Measurements made of the fuel and water consumption showed figures ranging from 1·47 to 1·04 lb. of anthracite per b.h.p.-hour and from 3·61 to 0·73 gallons of water per b.h.p.-hour. The enormous variation in the quantity of water required was very striking, and it showed that there was a considerable difference in the manner of operation of the various plants. As the water required for steam making is very small, practically the whole of the above difference

must have been due to the different quantities taken by the scrubber.

The Judges published **the following conclusions** as a result of the consumption trials—

That with a good suction producer plant, working continuously, at the specified loads and under the best conditions the following results may be anticipated :—

*With Anthracite.*

Full load : 1·1 lb. per b.h.p.-hour including fuel needed for starting, and for banking during the night.
Half load : 1·6 lb. per b.h.p.-hour including as above.
Water : 1 gallon per b.h.p.-hour at full load and ¾ gallon at half load.

*With Coke.*

Full load : 1·3 lb. per b.h.p.-hour including fuel needed for starting.
Water : 1½ gallons per b.h.p.-hour at full load.

Professor Dalby * also recorded as a result of these trials that—

" Assuming a 20 b.h.p. plant to start on Monday morning with an empty producer, and to run ten hours per day on full load for a week, banking the fires at night, the consumption of anthracite peas would be about half a ton for the week, and about ⅜ ton if the average load is about half full load. With **coke** the consumption is about **25 per cent. more.** From 2,000 to 3,000 gallons of water per week are required for a 20 b.h.p. plant to provide water for the scrubber and the producer, and of this by far the larger part would be used in the scrubber."

Tests were also made of the times taken to start up and to change load. As a result of their investigations the Judges awarded the premier places to the National and Crossley plants. The Judges were Professor Dalby and Capt. Sankey, R.E.

**88. Test of a Dowson Suction Gas Producer Plant.**—The following account of tests on two Dowson Suction Plants is

* B. A. paper, August, 1906.

extracted from Mr. Dugald Clerk's 1904 "James Forest" Lecture before the Institution of Civil Engineers. The tests were carried out by Mr. M. Atkinson Adam, B.Sc., Assoc. M. Inst. C.E. The first plant was adapted for a working load of 40 b.h.p. and the second for 30 b.h.p. In each case the producer was started up cold, and run on test for fully eight hours. At the start air was blown in by a small hand-power fan and after ten minutes from lighting up the gas was of a proper quality. The gas was then sucked through by a fan which represented the action of a gas engine operating under a constant load sucking gas from a producer in the usual way. Thence the gas passed to a gas holder. Analysis samples were frequently taken and the anthracite analyses were undertaken by Mr. Bertram Blount F.I.C., Assoc. Inst. C.E., whilst the gas analyses were carried out by Mr. Horatio Ballantyne, F.I.C. The heat efficiency of the producers was found in two ways :—

(1) Counting in the fuel used in the starting up operation which includes that necessary for the heating up of the plant.

(2) Omitting the first two hours of the test, and so giving the plant what may be termed a "flying start."

The quantities of water used are very interesting. The figures showed that for vaporization, the 40 b.h.p. plant used about 30 lb. per hour, whilst the 30 b.h.p. plant used about 20 lb. per hour. For the scrubber, the 40 b.h.p. plant used about 400 lb. per hour, and the 30 b.h.p. plant used about 380 lb. per hour. This shows how small a proportion of the total water consumption is needed for vaporization. The anthracite used was of an ordinary commercial kind, costing 14*s.* 6*d.* per ton at the pit, and about 24*s.* per ton delivered at Basingstoke. The efficiency figures for the two producer plants were found to be

| | | |
|---|---|---|
| "Standing start" . . | 40 b.h.p. | 85 per cent. |
| | 30 b.h.p. | 75 per cent. |
| "Flying start" . . | 40 b.h.p. | 89 per cent. |
| | 30 b.h.p. | 86 per cent. |

Reference should be made to the paper for detailed figures, but it may be mentioned that the gas was found on a general

average to have a calorific value of 135 B.T.U. per cubic foot, and have a composition as follows :—

| | | |
|---|---|---|
| $H_2$ | 5·5 | per cent. |
| $CH_4$ | 1·2 | ,, ,, |
| $CO$ | 20·0 | ,, ,, |
| $CO_2$ | 7·0 | ,, ,, |
| $O_2$ | 0·5 | ,, ,, |
| $N_2$ | 55·8 | ,, ,, |
| | 100·0 | |

**89. Tests of Pressure Producers.**—In 1904 some exhaustive tests were made in America on the results of employing different varieties of bituminous coal in pressure producer plants and in steam engines, and it is worth while to give a brief account * of some of the figures obtained.

| Kind of Coal. | Name of Sample. | Coal burned per E.H.P. | | Ratio of Coal used by Steam Plant to that used in Gas Plant. |
|---|---|---|---|---|
| | | Steam Plant. | Gas Plant. | |
| | | Lb. | Lb. | |
| Bitumin . . . | Alabama, No. 2 . . . . | 4·08 | 1·64 | 2·49 |
| Black lignite . . | Colorado, No. 1 . . . . | 4·84 | 1·71 | 2·83 |
| Bitumin . . . | Illinois, No. 3. . . . | 4·34 | 1·79 | 2·42 |
| ,, . . . | ,, ,, 4. . . . | 4·80 | 1·76 | 2·73 |
| ,, . . . | Indiana, No. 1 . . . | 4·13 | 1·93 | 2·14 |
| ,, . . . | ,, ,, 2 . . . | 4·35 | 1·55 | 2·81 |
| ,, . . . | Ind. Terr., No. 1. . . | 4·04 | 1·83 | 2·21 |
| ,, . . . | ,, ,, 4. . . | 4·64 | 1·43 | 3·24 |
| ,, . . . | Iowa, No. 2 . . . | 4·95 | 1·73 | 2·86 |
| ,, . . . | Kansas, No. 5 . . . | 3·93 | 1·62 | 2·43 |
| ,, . . . | Kentucky, No. 3 . . . . | 4·22 | 1·91 | 2·21 |
| ,, . . . | Missouri, No. 2 . . . | 4·93 | 1·71 | 2·88 |
| ,, . . . | W. Virginia, No. 1 . . | 3·90 | 1·57 | 2·48 |
| ,, . . . | ,, ,, 4 . . | 3·62 | 1·29 | 2·80 |
| ,, . . . | ,, ,, 7 . . | 3·55 | 1·46 | 2·43 |
| ,, . . . | ,, ,, 8 . . | 3·63 | 1·78 | 2·04 |
| ,, . . . | ,, ,, 9 . . | 3·46 | 1·40 | 2·47 |
| ,, . . . | ,, ,, 12 . . | 3·53 | 1·50 | 2·35 |
| ,, . . . | Wyoming, No. 2 . . . . | 5·90 | 2·07 | 2·85 |
| | | Average | | 2·57 |

* *The Times* Engineering Supplement, January 23, 1907.

In each case the output was about 200 e.h.p., and in most cases the length of the trials was from 10 to 30 hours.

Mr. Shober Burrows has reported the result of a 24 day test undertaken in 1906 on a **pressure producer plant** operating with bituminous fuel. Analysis of the fuel showed—

| | |
|---|---|
| $H_2O$ | 14·68 |
| Volatile combustible | 30·98 |
| Fixed carbon | 42·93 |
| Ash | 10·08 |
| S | 1·33 |
| | 100·00 |
| B.T.U. per lb. | =12,343. |

The gas left the generator at about 644° F. and passed a water seal to the scrubber. Thence to a centrifugal tar extractor. The calorific value of the gas was found to be 156 B.T.U. per cu. ft. and its composition was

| | |
|---|---|
| $CO_2$ | 9·2 |
| Ethylene | 0·4 |
| CO | 20·9 |
| $H_2$ | 15·6 |
| Methane | 1·9 |
| $N_2$ | 52·0 |
| | 100·0 |

About 143 lb. of tar was extracted per ton of coal used in the producer, whilst the approximate figures show that an average of 1·39 lb. of coal was used per b.h.p.-hour. As this plant ran for 24 consecutive days without shutting down, it is evident that continuity of operation can be practically achieved.

The whole of these tests go to show the great fuel economy obtained by the use of gas plant as contrasted with steam plant. Another feature in which the gas plant has the advantage is in the smallness of the **stand-by losses.** When a boiler is banked up for the night it consumes a very much larger quantity of coal during the period of banking than a producer plant of the same output would require. Actual measurements of this nature are recorded by Mr. Dowson

in his book on Producer Gas, and as a result it was found that in the case of steam power, the consumption of fuel per standing hour was 71·5 lb., and in the case of gas power, 3·5 lb. only, which shows a ratio of about 20 to 1. And since each of these figures is the mean of several tests, they are not open to the criticism that they represent isolated cases only.

It will be of advantage to record at this point what are the chief **objects to be achieved in the design and working** of producer plant—

(*a*) A fairly deep fuel bed should be allowed for, otherwise the air may blow through in thin places, and so lead to local variations in the temperature.

(*b*) Provision of some sort must be made to prevent caking or cavitation of the fuel.

(*c*) Fuel must be fed in and ashes removed in such a way as not to render the process discontinuous or intermittent.

(*d*) Leakage of gas from pressure producers must at all costs be avoided, as the gas contains a large proportion of poisonous CO.

There are a good many makes of pressure producer plant, and some are adaptable for by-product recovery. Among the latter one of the most prominent types is the Mond producer, which is being used on so large a scale in South Staffordshire. Here ammonia in the form of ammonium sulphate ($Am._2SO_4$) can be produced as a by-product and sold for a considerable amount—often more than enough to pay the coal bill. In this process, as has already been explained, the temperature of the producer must be kept low, and to do this, large quantities of steam are used, as much as $2\frac{1}{2}$ lb. of water per lb. of coal. This has the effect of course of reducing somewhat the actual efficiency of the gas producer and of raising the percentage of hydrogen present, but not to such a point as to introduce trouble in the engine.

**90. Percentage of Hydrogen.**—The percentage of hydrogen present in the gas to be employed in a gas engine regulates the amount of compression which can be used. A good compression is essential for high efficiency, but

if the proportion of hydrogen is over 30 per cent. the danger of pre-ignition has to be looked out for. The following table taken from a paper by Mr. J. R. Bibbins* shows the proportion of hydrogen present in various kinds of gas and the calorific value of the gas when taken alone, and when taken with its theoretically requisite proportion of air—

| Gas. | B.T.U. per cu. ft. | | $H_2$—per cent. by volume. |
|---|---|---|---|
| | Gas. | Mixture.† | |
| Natural Pittsburg | 978 | 91·0 | 3·0 |
| Oil | 846 | 93·0 | 32·0 |
| Coal-gas | 646 | 91·7 | 46·0 |
| Carburetted water gas | 575 | 92·0 | 40·0 |
| Water gas | 295 | 88·0 | 48·0 |
| Producer, hard coal | 144 | 68·0 | 20·0 |
| ,, soft | 144 | 65·5 | 10·0 |
| ,, coke | 125 | 63·0 | 10·0 |

Attempts have been made to reduce the proportion of hydrogen by admitting some of the **exhaust gases** into the producer instead of water vapour. In this case the dissociation of $CO_2$ replaces that of $H_2O$. This process is called the "straight carbon-monoxide gas producer." It is claimed to work very well and to permit of very high compressions being used. The gas has a calorific value of 105 B.T.U. and a composition of :—

| | | |
|---|---|---|
| $CO$ | 26·95 | per cent. |
| $H_2$ | 0·20 | ,, |
| $CO_2$ | 1·75 | ,, |
| $CH_4$ | 0·50 | ,, |
| $N_2$ | 69·30 | ,, |
| $O_2$ | 1·30 | ,, |

**91. Comparison of Costs.**—The following interesting com-

* "Fuel Gas for Internal Combustion Engines," *Cassier's Magazine*, 1906.

† Based on theoretical air for combustion.

parison has been drawn up by Mr. L. Andrews * and is well worth study.

CAPITAL COST OF 16,000 K.W. PLANT.

| | Steam Turbines. | Gas Engines. |
|---|---|---|
| | £ | £ |
| Engines and electric generators . . | 96,000 | 161,700 |
| Boilers, feed-pumps, coal, handling plant, etc. . . . . . . . . . | 81,000 | — |
| Producers, gas-cleaning and coal handling plant, with all pipes. . . . | — | 77,700 |
| Engine-room, building, cranes, and engine foundations . . . . . . | 18,000 | 42,000 |
| Switch-gear and wiring for ditto . . | 5,250 | 5,250 |
| | £200,250 | £286,650 |
| Allowance for contingencies, 5 per cent. | 10,012 | 14,332 |
| | £210,262 | £300,982 |
| Capital cost per K.W. installed . . | £13·1 | £18·88 |

RUNNING COST ON 100 PER CENT. LOAD FACTOR. ANNUAL OUTPUT = 140,000,000 K.W. HOURS.

| | Steam Turbines. | Gas Engines. |
|---|---|---|
| | £ | £ |
| Fuel, 165,000 tons at 10*s*. . . . . | 82,500 | — |
| Fuel, acid, stores and repairs for producers, less sale of by-products . . | — | 28,250 |
| Labour . . . . . . . . . . . . | 7,000 | 9,000 |
| Repairs of turbine plant, including boilers, etc.. . . . . . . . . . | 8,750 | — |
| Repairs of gas plant (excluding producers) . . . . . . . . . . | — | 6,000 |
| Oil, waste, and stores (excluding producer stores) . . . . . . . . | 1,750 | 4,370 |
| Interest and depreciation at 10 per cent. | 21,026 | 30,098 |
| | £121,026 | £78,118 |
| Total cost per K.W. hour . . . . | 0·204*d*. | 0·135*d*. |

* *Electrical Engineering*, October 24, 1907, and *S.A.*, January 30, 1908.

Mr. Andrews also takes the case when the load factor is only 15 per cent., and in that condition of running the costs per K.W.-hour came out at 0·545*d.* for steam turbines, and 0·566*d.* for gas engines. These rates are nearly the same, but with rise of load factor the balance would soon turn in favour of the gas plant. The Author considers Mr. Andrew's estimate of the capital cost of the gas plant, viz. nearly £19 per K.W., unduly high.

**92. The use of Gas Plant for Marine Propulsion** has been recently discussed before several engineering societies.

Mr. J. T. Milton in his 1906 paper before the Institution of Civil Engineers stated that he was led to give attention to engines of this kind in connexion with proposals to fit them in vessels classed with Lloyd's Register. The paper deals with engine problems only, and assumes that a proper and suitable type of producer capable of using cheap fuel will before long be available. The writer of the paper specifies the following conditions which must be satisfied by a successful marine engine—

(*a*) The engine must be reversible.

(*b*) It must be capable of being stopped quickly and of being started quickly either ahead or astern.

(*c*) It must be capable of being promptly speeded to any desired number of revolutions between dead slow and full speed, and of being kept steadily at the required speed for any length of time. "Dead slow" ought to be not faster than one-quarter of full speed, and should be less than this in very fast vessels.

(*d*) It must be capable of working well, not only in smooth water, but also in heavy weather, in a seaway in which the varying immersion of the propeller causes rapidly changing conditions of resistance.

(*e*) All working parts must be readily accessible for overhauling, and all wearing surfaces must be capable of being promptly and readily adjusted.

(*f*) The engine must be economical in fuel, and especially so at its ordinary working speed.

Certainly no existing engine complies with all these con-

ditions, and reference should be made to Mr. Milton's paper for a discussion of the difficulties : some curves are there given showing the different turning moment curves for different arrangements of engine.

Another paper is that read by Mr. J. McKechnie before the Institution of Naval Architects in 1907. Its title is "Propelling and Ordnance Machinery of Warships," and a portion of it deals with gas engine propulsion. It is stated that at the Vickers Works at Barrow-in-Furness there have been constructed internal combustion engines of a power equivalent to about 40,000 i.h.p., and that for three or four years almost continuous research work has been undertaken. As a result of the experiments a 2-stroke engine has been adopted. This engine, it is claimed, can be worked by producer gas, heavy oil, or compressed air, is reversible, and can take gas direct from a pressure producer without any scrubbing being necessary. To prevent the poisoning of the crew by the leakage of the gas from defective joints the pipes are jacketed with air under compression.

Not only would the introduction of gas engines for warship propulsion lead to a gain of space and dead weight (so allowing the offensive or defensive matériel to be added to), but the better disposition of its parts, and the absence of funnels would admit of a great improvement in respect of an actual increase in the number of guns which could fire on either broadside. In the proposed plan of battleship construction, the gas producers are shown divided into two sets well on either side of the ship, and the propelling machinery is shown well aft. The deck is clear for gun barbettes. Mr. Milton gives the following comparative table illustrating the superiority of the gas engine plant so far as area occupied, weight and fuel consumption are concerned—

Comparison of Weights, etc., of Steam, Gas and Oil Machinery for 16,000 h.p. Battleship.

| | Steam Engine. | Gas Engine. | Oil Engine. |
|---|---|---|---|
| I.H.P. available for propelling the ship . . | 16,000 | 16,000 | 16,000 |
| Weight of machinery, including usual auxiliaries, but not deck machinery. . . . | 1,585 tons* | 1,105 tons† | 750 tons‡ |
| I.H.P. per ton of machinery . . . . | 10·1 | 14·48 | 21·33 |
| Area occupied by machinery: Engines and boilers or producers . | 7,250 sq. ft. | 5,850 sq. ft. | 4,100 sq. ft. |
| Area per I.H.P. . . . | 0·453 sq. ft. | 0·366 sq. ft. | 0·257 sq. ft. |
| Fuel consumption in lb. per I.H.P. hour— | | | |
| At full power . . . | 1·6 lb. | 1·0 lb. | 0·6 lb. |
| At about ¼ full power | 1·66 lb. | 1·15 lb. | 0·75 lb. |

A further paper is Mr. A. Vennell Coster's before the Manchester Association of Engineers, dated 1907. Mr. Coster was fifteen years with the marine steam engine, three years at sea with the P. & O. and eleven years with Messrs. Crossley Bros., so that his experience gives his conclusions no little authority. The following are the advantages claimed for the gas engine—

1. The ship driven with half the amount of fuel.
2. Standard losses reduced over 75 per cent.
3. Working pressure confined to the engine cylinders.
4. No boiler tubes or main steam pipes to burst, nor furnace crowns to collapse.
5. No priming in a heavy seaway, or water hammer in pipes and cylinders.
6. No more difficulties with the firing of boilers on a beam sea. Gas producers may be charged only twice every twenty-four hours and the rolling and pitching of the vessel is rather an advantage than otherwise in assisting the fuel down from the charging hoppers.

* Includes water in boilers.
† ,, ,, jackets and piping, but not coal in producers.
‡ ,, ,, jackets and piping.

The three main difficulties in the way are—

(1) The construction of a gas producer able to gasify all grades of bituminous coal.

(2) A simple method to cleanse the gas from tar, either before the introduction of the fuel into the producer proper; when in the producer; or after the gas has left the producer on its way to the engine.

(3) Perfect control of the gas-propelled vessel in starting, stopping, reversing and running at all speeds.

The first of these difficulties obviously is avoided if coke or anthracite is used in the producer, but this solution is neither economical nor satisfactory on other grounds. Bituminous coal must be regarded as the source of the power to be used for ship propulsion. Mr. Coster stated that in his scheme for the cargo vessel *Lord Antrim* the producers were worked by means of a down-draught at the top, and an up-draught at the bottom, which met at the centre and the gas was drawn off by suction. The gas was then thoroughly sprayed and cleaned by being passed through coke, sawdust and wool wood scrubbers.

The reversing difficulty can be met in small engines by the use of a reversible propeller, but for obvious reasons this would not do in the case of large engines. For powers up to 500 h.p. gearing may be introduced to effect a reversal in the direction of propeller relation, just as in a motor car, but this cannot be used when the power transmitted is really large.

One of the great difficulties in connexion with the utilization of the gas engine on board ship lies in the fact that when the speed of the ship is decreased, the resistance to motion is decreased at a far greater rate, and this means that the mean effective pressure on the piston must be capable of very considerable reduction. When an attempt is made to get very low mean effective pressures in a gas engine, the engine is liable to stop altogether—in fact the gas engine as at present devised is **not sufficiently elastic** in its manner of working to make it an effective rival to the steam engine for marine purposes. The difficulty may be solved by driving generators from the gas engines, so pro-

ducing electric current which can be used in motors driving the screw propellers, but this requires a great weight of machinery, and is costly.

**93.** The well-known firm of **Thornycroft** have been working a good deal at the problem of adapting gas engines to ship

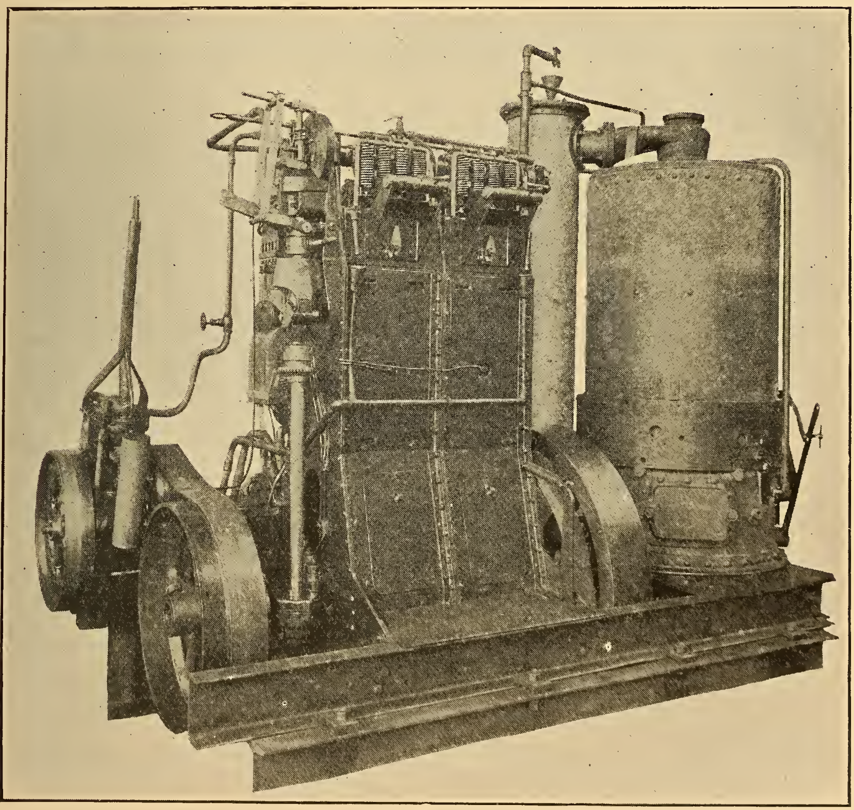


Fig. 57.—Two-cylinder Suction Gas Engine and Producer for Marine Purposes. (By courtesy of Messrs. J. I. Thornycroft & Co.)

propulsion, and illustrations are shown in Figs. 57 and 58 of the engines they are interested in. The chief difficulty is to devise a suction producer which will work with bituminous or caking coal without the necessity of being provided with apparatus for the extraction of tar and other by-pro-

ducts. The tar often amounts to 4 or 5 per cent. and may be as high as 15 per cent. It is therefore necessary to arrange the producer so that all the tar produced is consumed before it leaves the producer. This can be done by feeding in the fresh fuel from below, so that the heavy hydrocarbons given off from it are consumed as they rise into the hotter part of the fire. To save weight and space Herr Capitaine has hit on the idea of cleaning the gas by introducing a fine water spray, which mixes with the dust and other impurities, making a kind of fog. This fog then passes into a centrifugal machine which is driven fast enough to throw out the impurities and leave clean gas in the middle, which is then drawn off by the engine. Mr. J. E. Thornycroft * has given the composition of such gas as follows :—

| | |
|---|---|
| $CO_2$ . . . . . . . . . . | 6 per cent. |
| $CO$ . . . . . . . . . . | 25 per cent. |
| $CH_2$ . . . . . . . . . . | 1 per cent. |
| $H_2$ . . . . . . . . . . | 14 per cent. |
| $N_2$ . . . . . . . . . . | 54 per cent. |
| | 100 |

He also remarks that "it will be realized that the size of the producer for a given power is comparatively small when it is known that the area of the fire grate necessary is only 0·05 sq. ft. per h.p., whereas the average for an ordinary natural-draught steam boiler, burning 15 lb. coal per sq. ft. grate area, would be 0·2 sq. ft. per h.p."

The following test result is recorded by Mr. Thornycroft :—"Tests were made on November 8, 1904, with the *Gastug* No. 1 and *Elfriede*, a steam tug of very nearly the same dimensions and power. The *Gastug* No. 1 is 44 ft. 3 in. long by 10 ft. 6 in. beam, and is fitted with one of the four-cylinder 70 h.p. suction gas plants. The *Elfriede* is 47 ft. long by 12 ft. beam, and is fitted with a triple-expansion steam engine developing 75 h.p. At the towing meter the *Gastug* No. 1 attained a maximum pull of 2,140 lb., and the *Elfriede* a maximum of 2,020 lb. A run from

* Paper on "Gas Engines for Ship Propulsion," read April 5, 1906.

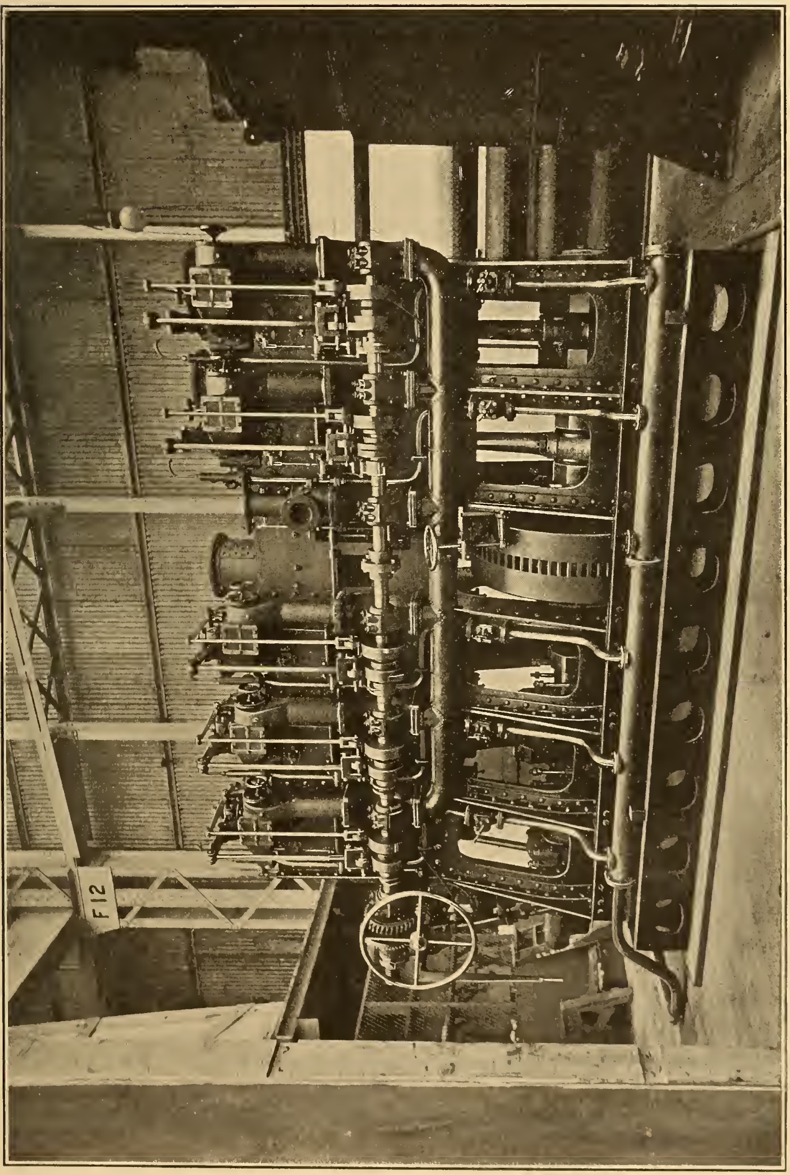



FIG. 58.—500 H.P. Suction Gas Engine for Marine Purposes. (By courtesy of Messrs. Wm. Beardmore & Co.)

Hamburg to Kiel and back was made by these two boats, during very stormy weather, at a maintained speed of $8\frac{1}{2}$ knots. The consumption of fuel was measured for a period of 10 hours, and was as follows—For the *Gastug* No. 1, 530 lb. of German anthracite : for the *Elfriede*, 1,820 lb. of steam coal. This shows an economy of 1 to 3·44 in favour of the gas plant."

## EXAMPLES.

1. Suppose that for 1·2 lb. of coal we get 1 brake horse-power-hour from a gas engine using Dowson gas. This works a reversed heat engine taking in heat $h$, in air at 10° C., and giving out heat $H$ at 20° C., the mechanical efficiency of the engine being 85 per cent. Find $H$ per lb. of coal and compare it with direct heating. The calorific power of the coal being 8,200 Centigrade heat units. (B. of E., 1899.)

2. Describe any non-luminous gas-making plant for use with a gas engine, working to, say, 100 indicated horse-power. What chemical actions take place in the gas manufacture ? What is the composition of the gas ? (B. of E., 1899.)

## APPENDIX A

The following is a description of the operation of a typical suction producer plant.

The suction type of gas-producing plant in question (**Campbell**) consists essentially of two main elements, a gas producer and a gas scrubber. In addition to these there is a simple form of separator through which the gas passes on its way from the producer to the scrubber, and in which it deposits the heavier particles of dust which are carried over from the producer. A gas box is also provided between the scrubber and the engine, to act as a reservoir, from which the engine can draw a regular supply of gas.

1. *Method of Gas Production.*—In this method of gas production air and steam at atmospheric pressure are drawn through incandescent fuel by the motion of the engine, the oxygen, hydrogen and carbon combining in the producer to form a combustible gas which is suitable for power purposes. No boiler for providing steam under pressure is required and no gasometer, the engine generating its supply of gas by the motion of the piston in the cylinder. The fuel used must be anthracite coal or coke

(bituminous coal must not be used). The steam is generated in an evaporator which is heated by the fire in the gas producer. The air is drawn into the producer over the surface of the heated water in the evaporator and in passing takes up the steam which it then carries through the producer.

2. *General Instructions.*—The coal used should be passed through a sieve and no pieces under $\frac{1}{4}$ in. (5 mm.) should be used. The most suitable size is $\frac{5}{8}$ in. to 1 in. (16 mm. to 25 mm.). Coal dust is not only of no value, but it tends to stop up the pipes and interfere with the working of the plant. The fuel should not be moistened before it is used. All the moisture required should be provided in the form of steam and pass through the fire in the ordinary way as described below. The evaporator should always be kept full of water to the overflow pipe. A water supply must be provided for the evaporator and the coke scrubber, and a drain to carry away the water from the scrubber. In any installation of this type, the engine should be erected as close to the producer as practicable so that the connecting pipes between the two are as short and direct as can be arranged.

3. *To start the Gas Producer after Erection or Cleaning.*—Before starting it is of the greatest importance to see that all the piping, cocks, and various vessels which go to make up the gas-producing plant *should be air tight*, as the apparatus when in operation is subjected to an excess of atmospheric pressure from without. If the various parts of the plant are not air tight, the air which leaks in will interfere with the quality of the gas and make it poorer. For this reason the whole apparatus should be tested after erection to prove the soundness of the joints, the test being carried out as follows: Referring to the illustration, if all the openings are closed except the cock *B*, and air is then blown into the apparatus by the hand fan *A* the various joints can be tested with a light. If air or gas escapes from the joints it will be at once detected. This test should be made periodically to see that everything is in order. It may be carried out at any time after cleaning, and when everything is proved to be in good working order the engine should be made ready for immediate use when the gas is available.

Provided that all the joints are sound and tight, the water should now be turned on to the coke scrubber by means of the tap *C* until it overflows through the pipe *D* provided for that purpcse. It is essential that the scrubber should contain sufficient water to seal the gas inlet.

Water should then be admitted to the evaporator *F* by means of the tap *G* until it just overflows in drops by the pipe *T* provided for that purpose. This overflow should be very slight before starting and must be regulated from time to time when

running according to the load on the engine as described below. The taps *C* and *G* and the cock *H* should now be closed. The cock

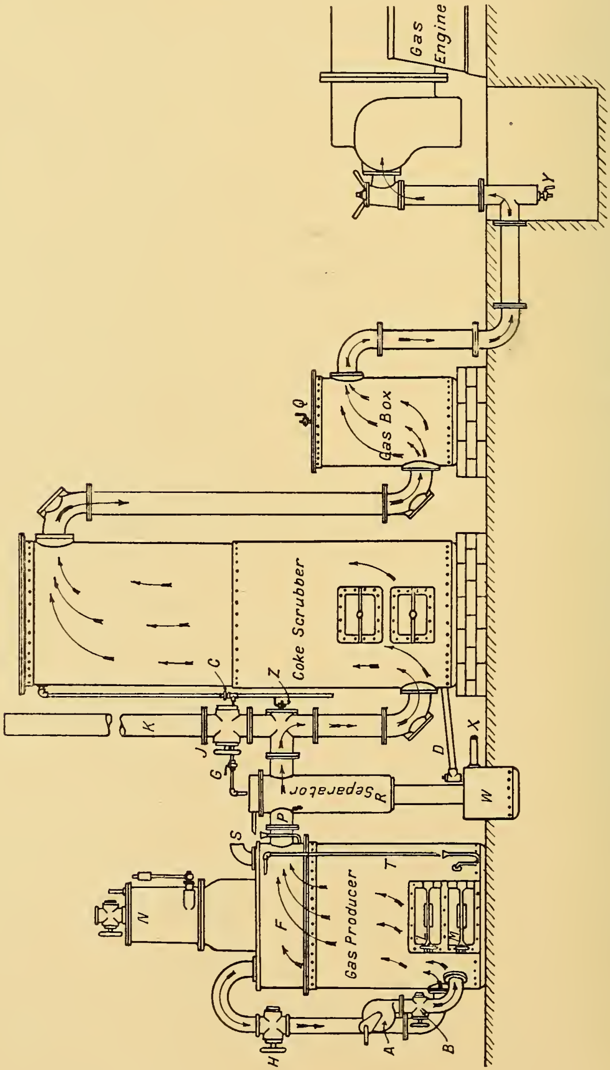



FIG. 59.—Diagram of Campbell Suction Gas Producer.

*B* and the cock *J* on the waste pipe *K* should be opened. The fire door *L* and the ashpit door *M* should then be opened and a fire of wood or coke started in the gas producer. *Ordinary*

*bituminous coal must on no account be used.* When the fire is burning up well anthracite coal should be added through the hopper $N$ in small quantities from time to time as the whole mass of fuel becomes incandescent throughout, this being continued until the gas producer is full to the level of the bottom of the gas pipe $P$. The fire door $L$ and ashpit door $M$ should be closed as soon as the coal is well alight. The hand fan $A$ must be used for the purpose of blowing up the fire when starting, the whole of the products of combustion being blown by its means through the gas pipe $P$, separator $R$ and uptake pipe $K$ to waste, the cock $J$ being open during this operation. Supposing the fire to have been lit for, say, fifteen to twenty minutes, and the hand fan to have been in operation during that time, the quality of the gas which is being made can now be tested by partially closing the cock $J$ and thus passing the gas through the scrubber and gas box to the test cock $Q$. The nearer this test cock is placed to the engine the better; it can be placed at any convenient point in the gas pipe, between the engine and the gas box, for example. Before passing the gas through the scrubber the water must be turned on to the scrubber, by the tap $C$. The blowing will have to continue for a few minutes until the scrubber and gas box are cleared of air, and gas has been blown in to take its place. A light should then be placed to the test cock $Q$ and the gas if of a good quality will burn with a steady flame. If the coal is of good quality the gas will burn with a long flame, orange red in colour, and one which does not go out. With some coals it is difficult to produce anything but a blue flame, but as long as the gas burns steadily it will generally be found that it is of sufficiently good quality to start the engine.

*Caution.*—When testing the gas, as described, care must be taken to turn the fan at a steady and even speed. Under no circumstances should the fan be stopped while the gas is burning at the test cock or the pressure will at once fall and the flame will probably be drawn back into the gas box and fire the gas in the gas box and scrubber, the explosion caused thereby blowing the water out of the water seal and possibly doing other damage. On the other hand the fan must not be blown too hard or the gas will be forced out through the water seal at the bottom of the coke scrubber.

The tap $C$ should be opened to such an extent that the temperature of the lower portion of the scrubber does not rise above 100° Fahr. approx., the top of the scrubber being cold. The amount by which the tap $G$ is opened must be regulated according to the load on the engine and so that the evaporator is always full and a slight surplus of water runs in drops only through the overflow pipe $T$ into the ashpit when the engine is running under a full load. When running under a light load

little or no water is required in the ashpit. An excess of water in the ashpit results in a poor quality of gas.

4. *To Start the Engine.*—As soon as the gas is burning satisfactorily at the test cock this cock and the cock *J* should be closed and the fan stopped. The engine should then be started in the usual way. No time must be lost in getting the engine started or the fire in the producer will become dull and a poor quality of gas be given off. Assuming that the engine has been started, the cock *H* should be opened and more water turned on to the scrubber by the tap *C* and to the evaporator by the tap *G*. The cock *B* should now be closed. By opening the cock *H* and closing *B* the air is drawn in through the inlet *S* and over the heated water in the evaporator *F*, the suction set up by the movement of the engine piston causing a constant indraught of air in the direction shown by the arrows in the sectional diagram. It may be mentioned here that the supply of air to the engine will have to be adjusted from time to time according to the quality of the gas. For this purpose a simple form of throttle valve should be provided in the air inlet passage through which air is supplied to the engine. This valve should be regulated so that as far as possible the engine takes in a supply of gas at every cycle and thus keeps the fire in the producer bright and in good condition. The engine should be provided with mechanism to ensure this being done.

5. *Method of Stoking the Gas Producer.*—The gas producer is provided with a hopper at the top for the purpose of feeding the fire. The hopper is provided with a swing door at the top and a valve with a weighted lever at the bottom so that when fresh coal is added the top door only is opened, the valve remaining closed. When the coal has been filled in through the hopper the top door is closed and the valve opened. By this means all air is excluded from the gas producer. Care should be taken to see that the valve to which the weighted lever is attached is properly closed so that no air can enter while the gas producer is working. Generally speaking it will be necessary to add a charge of anthracite every two or three hours; this, however, must depend upon the size of the apparatus and the amount of power which the engine is developing. While the gas producer is in full operation the coal should not be allowed to fall below the lowest point in the evaporator. The top layer of coal should never be incandescent, this point can be watched through the mica window which is provided for that purpose at the top of the hopper. Previous to stopping the engine, however, the fire in the producer should be burnt down so as to leave only a moderate quantity of coal in the producer, sufficient to start up quickly again when required. How frequently the fire will have to be cleaned will depend

upon the quality and amount of the fuel used; speaking generally twice a day will be sufficient, once in the morning before starting and once at midday, if a stoppage is made then. Should it be necessary to stir up the fire whilst the engine is at work this can be done through a hole in the ash door by means of a poker. If it is necessary to take out the clinker whilst the engine is at work, this should be done very quickly so as to allow as little air as possible to enter the gas producer, for should an excess of air be allowed to enter, the gas would be of inferior quality. It is advisable as far as possible to leave the gas producer alone whilst the engine is at work, except for the occasional charges of coal which it requires. The gas producer should be cleaned out entirely about once a week and the clinker chipped off the firebrick lining of the producer if necessary. The producer should never be cleaned directly after the fire is raked out, but should be allowed to cool down gradually, otherwise the firebrick lining will probably crack through the rapid change in temperature.

6. *Hydraulic Box.*—The surplus from the coke scrubber is led into the hydraulic box *W* by means of the pipe *D*. This water forms at the same time a water seal for the pipe which connects with the separator mentioned above. The box *W* should be cleaned out every few weeks so as to keep it clear of the accumulated ash and small particles of coal which will come over with the gas. Special attention should be paid to the pipe *D* to see that no foreign matter settles in it. When the engine is at work the surface of the water in the hydraulic box will be in constant movement; the movement which should be a slight one, will vary with the amount of gas drawn away by the engine. An overflow pipe *X* is provided to run the water away from this box to a drain, or as may be arranged.

7. *Coke Scrubber.*—The coke scrubber is provided to remove from the gas all its impurities and at the same time to cool it. When the apparatus has been erected the inside of the scrubber should be thoroughly cleaned and the grating put in through the upper manhole. The scrubber should then be filled with well washed foundry coke, the size being not less than about 1 in. (25 mm.). The bottom layer of coke for a depth of about 8 in. (0·2 m.) should consist of pieces which are under any circumstances so large that they will not fall through the grating. The scrubber can then be filled up with coke to about 4 in. (0·1 m.) below the water pipe. Before starting the bottom of the scrubber should be cleaned out through the doors provided for that purpose. All the openings in the scrubber should now be closed and the water supply turned on so that the coke is washed thoroughly free from all the particles of dust which it may contain.

Every three or four weeks the bottom door of the coke scrubber should be opened to see whether there is any accumulation of dust in the form of mud at the bottom of the scrubber; this if present should be removed. When the coke is first put into place this examination should be made more frequently, as new coke frequently contains a large quantity of dust. The coke in the scrubber will, generally speaking, be serviceable for a period of nine to twelve months, but this depends upon the amount of work which the plant has to do. When it is found necessary to renew the coke in the scrubber the whole of the apparatus must be stopped, the waste pipe opened and all the ash and fire hole doors opened and left open for several hours before any work is done to the plant. The upper cover of the scrubber should then be removed and the coke taken out through the upper side door in the scrubber. This cleaning should take place during the daytime so that no fire or light need be brought into the gas-plant house while it is going on. The windows of the house should be open during the process of cleaning so that there is plenty of ventilation. It is advisable that there should always be two men present during the operation of cleaning, in case one of them should be overcome by the presence of gas. When replacing the doors on the scrubber after having renewed the coke care must be taken to see that the joints are sound and tight as already described.

8. *Piping and Gas Box.*—These should be looked to and cleaned about once a month. Impurities will settle in any pockets or where the course of the gas is not direct. For this reason all bent pipes should be avoided as far as possible and when present should be examined from time to time. The moisture which condenses in the gas box and in the pipe leading from it to the engine should be emptied out daily, otherwise it will get into the engine and interfere with its working. A drain cock should be provided, as at *Y*, for the purpose of drawing off this moisture.

9. *To Stop the Gas Producer.*—The gas cock on the engine should be shut and the waste cock *J* opened so as to allow the remaining gas to escape. The taps *C* and *G* and the cock *H* must then be closed and the ash door opened a few inches so as to allow the fire to continue burning.

10. *To start the Apparatus again after a Temporary Stoppage.*—The fire and ash doors should be opened to clean the fire, any cinders or clinker should be removed without disturbing the fire as far as this is possible, the doors should then be closed, the cock *B* opened and the fan started until the fire is again in good condition. Anthracite must then be added until a good quality of gas is obtained when the engine may be started up to work. When the stoppage is only temporary, the scrubber and gas box will probably be full of good gas when it takes place

and it is therefore better to test the fresh gas, made at restarting, by means of a test cock placed at $Z$ rather than to test it at $Q$. When good gas is obtained at $Z$ the cock $J$ can be closed and the gas then sent through the scrubber and gas box to the engine. By following this plan the good gas remaining in the scrubber and gas box when the plant was stopped will be utilized, instead of being blown away to waste as might otherwise have been the case.

*Caution.*—The regulation of the supply of water to the coke scrubber is important. If the supply be too small, steam will be formed in the scrubber, the gas will not be properly cleaned, and the quality of the gas will deteriorate. If the supply be too great, the water seal of the gas pipe will be too deep and the engine will not be able to suck the gas through the producer. The coal should not be too large or of unequal size or the air spaces between the various pieces will be too great. The guiding principle in this is to have a mass of fuel in the producer which is as homogeneous as possible without being solid. Where coke is used as the fuel a sawdust scrubber is required between the coke scrubber and the gas box. When a gas plant has been designed for anthracite, other modifications may be necessary if it is decided to change from anthracite coal to coke.

## CHAPTER VII

# Blast-Furnace and Coke-Oven Gases

Thermal Value—Cleaning the Gas—Utilization of the Surplus Power.

**94. The Production of Waste Power from Blast-Furnace and Coke-Oven Gases.**—The idea of using blast-furnace and coke-oven gases in gas engines is a relatively modern idea, and the extent to which it may be put into force in any country depends chiefly upon that country's output in pig-iron. The following figures show the output in pig-iron in metric tons for the three chief countries concerned—

| | 1905. | 1906. |
|---|---|---|
| U.S.A. | 23,340,258 | 25,712,106 |
| Germany | 10,987,623 | 12,478,267 |
| Great Britain | 9,746,221 | 10,311,778 |

The gas that issues from blast furnaces is rich in **carbon-monoxide** and poor in **hydrogen,** and has a calorific power of about 90 B.T.U. per cu. ft. : whereas the gas from coke ovens is extremely rich in hydrogen and may have a calorific value as high as 500 B.T.U. per cu. ft. The former is the easier to deal with as it comes in steadier quantities, and with the small quantity of hydrogen which it contains, pre-ignitions are not likely to occur. Consequently it is safe to raise the compression to a much higher point (180 lb. per sq. inch or more) than would otherwise be safe, and the engine is thereby rendered of higher thermal efficiency. Both gases require cleaning so far as dust is concerned.

**95. Blast Furnace Gases.**—The idea of burning blast-furnace gases directly in gas engines instead of under steam boilers, as had previously been done, was first put into practice about the year 1894, nearly simultaneously in Great Britain, Germany and Belgium. The pioneers, prominent among whom was the late Mr. B. H. Thwaite, experimented with small engines and, as satisfactory results were obtained, it was soon desired to increase the scale of operation. In Germany great progress has now been made and recently a number of large plants have been put in in this country and in the U.S.A.

The calculation as to the power available in this way in Great Britain may be made in the following manner. The pig-iron output for 1906 (for example) was, in round figures,

$$10{,}000{,}000 \text{ tons,}$$

and it is well established that the residual gases from blast furnaces in Great Britain as well as on the Continent and in America, are capable when used in internal combustion engines of yielding about **27 h.p. per ton of pig-iron per day** (the figures given by various engineers are as follows: Greiner, 20; Bryan Donkin, 28; Max Rotter, 25; Thompson, 20; Rossi, 30 to 35). It follows that the whole output would be about

$$\frac{10{,}000{,}000}{365} \times 27 = \mathbf{740{,}000 \text{ h.p.},}$$

of which at present the greater part is going to waste.

The corresponding h.p. for the 12,500,000 tons of output in Germany would be 930,000 h.p., which agrees generally with Dr. Hoffmann's estimate of 1,000,000 h.p.

It is confidently calculated that in those countries where this development is in progress a saving of several shillings per ton will be made in the cost of producing iron. Several German firms, notably, have already found very favourable financial results to accrue.

Professor H. Hubert remarks * that in Belgium the honour of being first in the field belongs to Messrs. Bailly and Kraft,

* Iron and Steel Institute, 1906.

of the Cockerill Co. The patent taken out by the Company for this new application was dated May 15, 1895, and the first trials were made at the end of that year. They were made with a Simplex engine of 8 h.p. in which the clearance space had been reduced in order to increase the compression and to facilitate the ignition of the mixture. The gas cleaning was imperfect, and was carried out simply by passing it through two scrubbers four metres high. The engine is stated to have displayed perfect elasticity, and adapted itself to the variations of composition, pressure and temperature of the gases.

The following interesting table is taken from Professor Hubert's paper—

| Engine. | Date of Trials. | Power. | | Calories used per I.H.P. Hour. | Thermal Efficiency. |
|---|---|---|---|---|---|
| | | I.H.P. | B.H.P. | | |
| | | | | | per cent. |
| 8 h.p. engine . . . . . . | 1896 | 5·26 | 4 | 4,030 | 15·77 |
| 200 h.p. engine (single cylinder, single acting, constant admission) . . . . . . . | 1898 | 213·9 | 181·82 | 2,775 | 22·9 |
| 600 h.p. engine (as above) . | 1900 | 825·8 | 670·0 | 2,520 | 25·2 |
| 200 h.p. engine (as above, except for variable admission). | 1901 | 246·9 | 215·3 | 2,766 | 23·0 |
| 1,400 h.p. engine (double-acting tandem, variable admission). | 1906 | 1,755 | 1,582 | 2,129 | 29·8 |

**96. Coke-Oven Gases.**—Coke-oven gases are much richer in hydrogen than blast-furnace gases, and they are therefore much more liable to pre-ignitions. To avoid this danger, the compression is not taken so high, although this precaution unfortunately has also the effect of tending to reduce efficiency. On the other hand their thermal value is far higher, often more than five times as high. To illustrate this, the following typical figures are given

B.F. gas :—24½ per cent. of CO ; 62 per cent. of $N_2$ ; 1¼ per cent. of $H_2$ ; Calorific value 86 B.T.U. per cu. ft.

C.Oven gas :—50 per cent. of $H_2$ ; 40 per cent. of $CH_4$ ; Calorific value 560 B.T.U. per cu. ft.

To calculate the possible output obtainable from coke-

oven gases in this country is not difficult. Taking the 1906 output of pig-iron as 10,000,000 tons, the consumption of hard coke may be put as about 11,000,000 tons. To produce this quantity of coke about 15,000,000 tons of coal would be required, which on coking would give off about one-fifth of its weight in the form of gas, corresponding to about 500,000,000 cubic feet of gas per day. Assuming that a quarter of this is available as a surplus for use in gas engines, and that it is of the thermal value of 500 B.T.U. per cu. ft., the corresponding thermal energy is easily calculated. If the gas engines used have a thermal efficiency of 30 per cent., the following h.p. would be available :—

$$\tfrac{1}{4} \times 500{,}000{,}000 \times \frac{500}{24 \times 60} \times \frac{778}{33{,}000} \times 0\cdot30 = 306{,}000 \text{ h.p.},$$

or in round figures **300,000 h.p.** This is an estimate for the English output. Dr. Hoffmann has estimated the German output as from 550,000 to 600,000 h.p. Not a little enterprise has been shown in Germany in harnessing this source of power, and action is being taken in this country to the same end.

The proportion of one quarter, used in the above calculation * as to the fraction of the gas available for the production of this surplus power, depends upon chemical problems, but it has recently been found that by raising the temperature of the air entering the ovens to 1,000 or 1,100° C. by means of regenerators, only 45 to 55 per cent. of the total quantity of gas evolved from the fuel is required for the work of heating the ovens, so that practically half the gas would in that case be available for the production of power in gas engines. This idea has been worked out by Mr. Koppers, and at the Anna Colliery of the Eschweiler Mining Co., near Aix-la-Chapelle, there are reported to be six batteries of Koppers regenerator ovens, with a power station designed for the production of 16,000 h.p. from the surplus gas.

* M. Léon Greiner gives the following approximate rules for the amount of surplus power available for use :—(*a*) with blast furnaces, the continuously available h.p. is equal to the number of tons of iron made per month ; (*b*) with by-product recovery ovens, the continuously available h.p. is equal to the number of tons of coke made per week.

It is on record* that at the Wath Main Colliery, Wath-upon-Dearne, Rotherham, an installation of 30 Hüessener patent by-product coke ovens, erected by the Coal Distillation Co. of Middlesbrough—representing the Actien Gesellschaft fuer Kohlendestillation—has been put in. The plant is to produce 800 tons of blast-furnace coke per week, and there is to be available sufficient surplus gas and surplus waste heat to produce 300 h.p. of electricity from the 30 ovens, in addition to meeting the requirements for power for coal grinding, elevating, and by-product plants. There are other instances of similar enterprise whereby English firms, on discarding the old "**beehive**" type of oven, have been able to obtain large quantities of surplus power. Of course there are other by-products besides power produced from coke ovens, such as sulphate of ammonia, coal-tar and benzole.

**97.** The Shelton Iron Works have some Koerting Engines working on **coke-oven gases,** and it has been found† that when some coals are used a calorific value of over 600 B.T.U. per cu. ft. is obtained, although 400 is more common. In no case, however, is the quality constant during the whole period of coking. It usually decreases from about 450 to 350 during the operation. The gas passes through scrubbers where the ammonium sulphate and other by-products are collected and most of the tar removed. The gas then is divided into two almost equal parts, one half going to heat the coke ovens, and the rest to the production of power. As the gas contains much hydrogen, naphthalene, and other highly inflammable bodies, it is liable to pre-ignitions, and the compression is kept down to 100 lb. per sq. inch. instead of the 140 lb. per sq. inch, which would otherwise be customary. The mean pressure works out at about 75 lb. per sq. inch. On testing a new variety of fuel the following results were obtained: Thermal value of gas 381 B.T.U. per cu. ft., engines developed 1 h.p. per hour per 22 cu. ft. at full load, or a thermal efficiency of $\frac{1,980,000}{22 \times 381 \times 778} = 0{\cdot}305$, which seems very high. The analysis

* T.E.S., April 17, 1907.
† *Engineering,* February 15, 1907.

of the gas was

| | |
|---|---|
| $CO_2$ . . . . . . . . . . . | 3·55 per cent. |
| Olefines, etc.. . . . . . . . | 5·18 ,, ,, |
| $O_2$ . . . . . . . . . . . | 1·59 ,, ,, |
| Methane . . . . . . . . . | 27·82 ,, ,, |
| $H_2$ . . . . . . . . . . . | 54·33 ,, ,, |
| $N_2$ . . . . . . . . . . . | 3·16 ,, ,, |

According to some figures in *The Engineer*, of 22 installations in Germany with a total output of 13,000 h.p. from engines working on coke-oven gas, no less than eleven, or half of them, do not find it necessary to clean the gas. One of them was stated to be using gas with 0·2 per cent. of sulphur without injurious effect on the iron.

**98. Cleaning the Gas.**—It has been found that the most effective way of cleaning the gas is by the action of a water fed fan. The gas passes through a **centrifugal fan** which

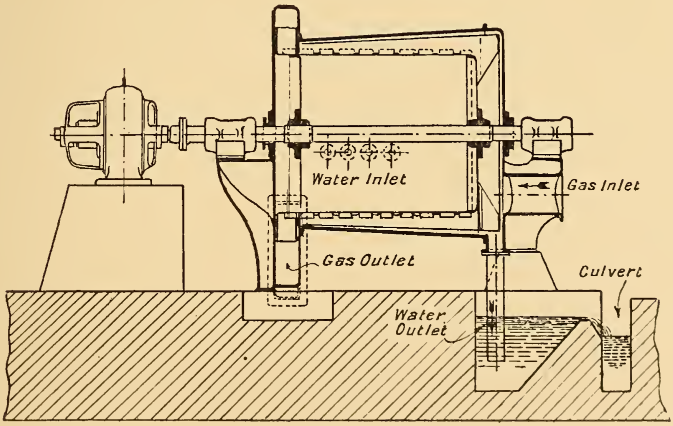



FIG. 60.—Theisen Gas Washer—Section.

causes the heavy particles of dust to fly outwards, and at the same time water is fed into the fan and broken up by the same centrifugal action. This water catches up the dust particles and passes with them to a sump. Perhaps the best known gas cleaner of this type is the Theisen Patent Cen-

trifugal Central-flow Gas Washer, made by Messrs. Richardsons, Westgarth and Co. It is illustrated in Figs. 60 and 61. The Theisen machines are specially adapted for cleaning gas, and particularly blast-furnace gas, for use in gas engines and where a high degree of purity is required. When very hot and dirty gas has to be treated, it is considered advisable to instal a preliminary saturator before the washer, where the gas may be cooled and the heavier dust removed. In this way not only is the volume of gas to be cleaned reduced, but less water is required in the washer itself, and consequently less power is absorbed. The makers claim that the

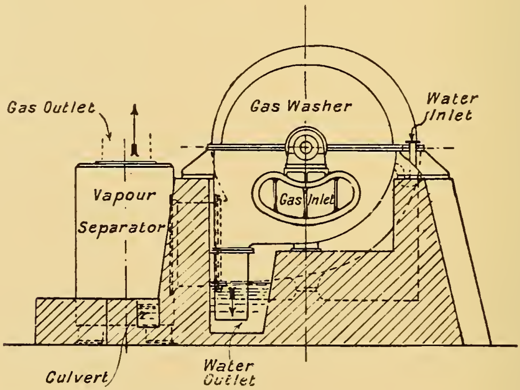



Fig. 61.—Theisen Gas Washer—End Elevation.

power taken to drive the cleaner does not exceed 2 per cent. of the maximum power which could be generated in gas engines from the gas cleaned. The quantity of water required by the Theisen apparatus varies with the temperature of the gas and the amount of dust therein, and in addition with the degree of cleaning necessary. With hot and dirty gas it sometimes happens that as much as 1 litre of water is required per cubic metre (or 1,000 litres) of gas cleaned, but usually half this quantity will suffice. Of course the water can be used again and again, if the dust be allowed to settle out of it. The makers have published the following table showing results of trials—

## Results Obtained with Theisen's Apparatus Cleaning Blast Furnace Gas.

| Name of Ironworks. | Quantity of Gas in Cub. met. per hour. | Particulars of Gas. | | | | | | Water Used. | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Entering. | | | Leaving. | | | Temperature. | | Quantity. | |
| | | Dust in grms. p. cu. met. | Temp. | Moisture, grammes p. cub. met. | Dust in grms. p. cu. met. | Temp. | Moisture, grammes p. cub. met. | Entering. | Leaving. | Gallons per hour. | Gallons p. cub. met. gas. |
| | | Hot Gas direct from Furnaces. | | | | | | | | | |
| Hochdahl | 17,200 | 6 | 219° F. | 17·8 | ·04 | 86° F. | 7 | 57° F. | 102° F. | 4,163 | ·243 |
| " | 12,000 | 6 | 316° F. | 24 | ·02 | 99° F. | 5 | 45° F. | 104° F. | 2,650 | ·22 |
| Schalke | 10,200 | 3–4 | 291° F. | 15 per cent. vol. | ·004 | 86° F. | 12–20 | 54° F. | 131° F. | 2,250 | ·22 |
| | | Cooled Gas with heavy dust separated. | | | | | | | | | |
| Ormesby, near Middlesbro' | 7,500 | 2 | 126° F. | 3 | ·0050 | 73° F. | 18·5 | 58° F. | 80° F. | 448 | ·06 |
| Hoerde | 12/15,000 | 2·5 | 115° F. | 32 | — | 91° F. | 3·45 | 82° F. | 99° F. | 2,650/3,530 | {·23 ·234} |
| " | 6,000 | 2·34 | 113° F. | 36·21 | ·01 | 82° F. | 3·013 | 68° F. | 93° F. | 1,540 | ·253 |
| Rombach | 9,000 | 2 | 109° F. | 42 | ·02 | 97° F. | 3·2 | 64° F. | 66° F. | 2,250 | ·25 |

Mem.—1 Cubic Metre = 35·3 cubic feet.

The **amount of dust** in the gas can be measured very easily. It is only necessary to pass the gas through a filter consisting of a glass tube filled with absorbent cotton. The quantity of gas passed is measured in a meter, and the cotton is weighed before and after. The method is stated to give accurate results if the cotton is evenly packed along the tube and is not hydroscopic. In any case the cotton should be dried before and after in a desiccator, and weighed from time to time to check whether any moisture is held in it.

In America,* peculiar difficulties are experienced, owing to the character of the ores used. The Mesati ores are stated to be especially troublesome, owing to their friable nature. With every disturbance in the furnace great quantities of dust are evolved, which often pass the entire cleaning plant unless unusual precautions are taken. The suspended matter consists largely of ore dust together with some additional matter carried over from the other constituents of the furnace charge. At Bessemer, however, a cleaning plant has been put down which has cleaned the gas as low as 0·1 grains per cubic foot, which is considerably cleaner than the surrounding air in that particular locality. In practice, however, the engines work quite well with ten times this amount of dust.

**99. Utilization of Surplus Power.**—The utilization of the power derivable from the waste gases of blast furnaces and coke ovens is a problem in itself. The solution of this problem must depend upon the extent to which local demand for power exists, or can be created. It is only necessary to think of such electro-metallurgical processes as the manufacture of **aluminium** to bring to mind the possibility of the creation of huge demands for current under the happiest conditions in respect of load factor. For the transmission of such power for any distance less than half a dozen miles, it would probably be most economical to use pipe lines to convey the gases, but for longer distances electrical transmission would be the obvious method to adopt. Another industry that might be served is the manufacture of **calcium carbide.** Carbide is not now being manufactured in bulk in this

* T.E.S., July 17, 1907.

country, owing to the lack of cheap power. Abroad, engineers have the advantage of extraordinarily cheap water power—as low, according to Professor S. P. Thompson, as $\frac{1}{26}$ part of a penny per h.p.-hour—and it is clear therefore that unless some very cheap source of power is rendered available here also it will not be possible for this country to produce its own carbide. Calcium carbide, in its purest form, is used for the production of acetylene for lighting purposes, but a less pure and cheaper kind can be used in the preparation of **chemical manure,** for which the demand is on an altogether larger scale.

Lime and coke when heated together to a temperature of 2,000–3,000° C. produce calcium carbide, combining in accordance with the following chemical formula—

$$CaO + 3C = CaC_2 + CO.$$

This reaction is carried out in an electric furnace worked either by direct or alternating current, although as the latter allows of a higher voltage transmission and simple transformation, it is usually preferred. It is a high temperature reaction and not an electrolytic one, thus permitting either type of current to be used. In the above equation the CO passes away as a by-product, and carries with it one-third of the carbon used. This gas might of course be collected and its thermal value used say for the heating up of the charge of lime and coke, for the earlier part of the great temperature range necessary. The amount of current needed to produce 1 ton of calcium carbide is about $\frac{2}{3}$ h.p.-years. Mr. Bertram Blount in his *Practical Electro-Chemistry* remarks :—" The surplus gas (from coke ovens and blast furnaces) can be used with economy in large gas engines of 500 or 1,000 h.p., and energy thus obtained almost as cheaply as from a water-power. For example, at an inclusive cost of $\frac{1}{10}d.$ per h.p.-hour, which is by no means unattainable, the price per h.p.-year is £3 13*s*., a figure which approaches that of a moderately cheap water-power. The real obstacle to the general utilization of such power is not its cost, but the somewhat restricted market for carbide, causing it to be readily swamped by any great increase of supply ; even with that restriction, however, the

manufacturer having cheap coke and lime in an industrial centre, will stand at least as good a chance as his rival with slightly cheaper power, but away from such supplies."

**100. Owing to the discovery** that calcium carbide could be used in the preparation of an excellent chemical manure, the possibility has been opened up of an enormous demand for this product, thus affording a suitable purpose to which large quantities of electric power could well be devoted. Such an enlargement of the calcium carbide market might not be altogether welcome to present manufacturers of the carbide, as the new product, not being used in the production of acetylene gas for lighting need not be so pure. A heavy demand for the less pure carbide might therefore lead to difficulty in obtaining small supplies of a purer kind, as it would hardly be worth while undertaking it. Or even if undertaken, the cost of such carbide might actually be greater with the increase of output than it is now. Probably if the bulk of the output were of a different quality it would not be feasible commercially to produce raw carbide in so pure a state, but this would not prevent the impurer carbide being purified by subsequent treatment in such quantities as the acetylene demand might necessitate. Even if chemical difficulties present themselves in the purification of the carbide when made, there is no reason to suppose that the ingenuity of chemists will be unable to circumvent those obstacles as soon as it is necessary for them to be dealt with. Present-day manufacturers hold to prevention being better than cure, and would far rather see that purer raw materials (coke and lime) were used; but if a big agricultural demand should arise, it is not to be expected that subsequent modes of manufacture would be controlled entirely with a view to the smaller market. The virtue of calcium carbide from the agricultural point of view lies in the fact that it can be converted into **calcium cyanamide,** which can be directly applied to land as a fertilizer, and that when so employed it is of great value and efficacy. The cyanamide can be obtained direct from the carbide by fusing the latter in a stream of nitrogen. Or if preferred, the process may be shortened by admitting nitrogen to the electric furnace in which the

lime and coke are being fused. As the author pointed out some years ago *—"The question for English engineers is, whether it is commercially possible (scientifically it certainly is), for a carbide industry to be set up in this country, drawing its **electrical power from the waste gases** of the English iron districts, and relying on a sufficient economy in regard to other items of cost, such as capital charges, labour, lime, coke, and carbon electrodes to enable carbide to be produced and sold at a profit at a rate not higher than the £13 or so a ton for which it can now be obtained in the market. It appears to the writer that this recovery of a British industry is well worth attempting, especially in view of the possible creation of an agricultural demand so keen that the supply would of necessity lag behind the demand for some years, thus providing a condition which would be most favourable to the initiation of the enterprise. What the saving on freight would be if the carbide were produced in this country instead of abroad would depend largely upon the relative views taken by railway and steamship companies as to its 'danger' in transit, but it is fairly obvious that the balance of the advantage, whatever it may amount to, should, in most localities, lie with the English producer."

As above quoted, Mr. Blount put the figure at which power could be obtained from waste gases at as low a figure as £3 13*s*. per h.p.-year, but Mr. B. H. Thwaite, in his Iron and Steel Institute paper in 1907, put the cost at an even lower figure than that. His figure was £3 6*s*. 8*d*. per K.W.-year, corresponding to less than £2 10*s*. per h.p.-year, and Mr. H. Greiner, who followed in the discussion, stated that his experience showed a cost of about 80 francs per K.W.-year, equivalent to about £2 8*s*. per h.p.-year. Mr. Thwaite's plan was to collect the waste gases from all the furnaces of the district in which iron was being made, and utilize it in gas engines for the production of electric power which would be distributed to customers. The first call on the former would be for the demands of the steel works concerned, and the balance could be sold to any one else who might want power. This plan would have the merits of decreasing capital charges

* T.E.S., October 3, 1906.

for plant, and of increasing the load factor. According to Mr. Greiner about 7,000 e.h.p. was then being generated at the Cockerill works from coke-oven and blast-furnaces gases, and it was intended to increase this output very largely. In the United States the utilization of blast-furnace gases in gas engines practically began at the Edgar Thompson works, of the Carnegie Steel Co. at Bessemer near Pittsburgh, by the installation of several 3,000 h.p. Westinghouse engines. The experiment being successful the United States Steel Corporation decided to increase the capacity of the gas engine plant to 50,000 h.p. It is reported that very little trouble has been found in the working of the plant so far installed, despite the absence of experimental data or experience of continuous working.

*SECTION III*

# OIL AND PETROL ENGINES

# CHAPTER VIII

# Oil and Petrol Engines

Fuels—Slow-Speed Oil Engines—Diesel Engine—Petrol Engines— Carburettors—Governors—Theory of Jet Carburettors—Ignition.

**101. Fuels.**—Internal combustion engines are of two classes: (1) those that work with gases for their explosive medium, and (2) those that use vapours of liquid hydrocarbons such as oils. The former class has been dealt with in the preceding chapters so far as everything except methods of ignition is concerned—and ignition being similar in both classes does not need to be dealt with in two parts. Oil and petrol engines, as those in class (2) are generally named, are of practically the same design as gas engines so far as cylinders, pistons, valves, etc., are concerned, and the difference between them mainly relates to the mechanism for dealing with the fuel used. A gas engine does not need any carburettor, whereas in the oil or petrol engine it is one of the most important and most sensitive parts.

Fuels for oil and petrol engines may be divided into (1) **heavy oils** and (2) **spirits.** Heavy oils include everything from crude Borneo oil (looking like thin treacle) to paraffin (such as is commonly used in oil lamps). The spirits used are chiefly petrol, benzol and alcohol, but a great number of other variations have been suggested by ingenious persons. The reason why it is so easy to find new petroleum spirits which are usable in internal combustion engines is that each change in the temperature range over which the oil is distilled practically produces a new substance. The names of the various spirits are therefore numerous and not a little confusing. There is the additional complication that in the United States what would here be called petrol is known as gasolene, and for paraffin they use the word kerosene.

The following are the calorific values of the various fuels named—

| Fuel. | Calorific Value (higher value) In B.T.U. per lb. | Calorific Value (higher value) In ft.-lb. per lb of Fuel. |
|---|---|---|
| Absolute alcohol . . . | 12,600 | 9,800,000 |
| Methyl alcohol (with density—0·82) . . . . | 11,300 | 8,800,000 |
| * Petrol with density = 0·722 | from 20,300 to 19,300 | from 15,800,000 to 15,000,000 |
| Paraffin . . . . . . . | 23,100 | 18,000,000 |
| Denatured alcohol (methylated spirits) density = 0·83 | 11,000 | 8,600,000 |

**102.** According to Professor Vivian B. Lewes, as the **crude oil** comes from the well it is a mixture of many hydrocarbons and varies considerably both as regards its physical and chemical properties, according to the source from which it was obtained; the American and Russian oils upon which most scientific work has been done differ widely in their chemical composition, although much alike in their physical properties.

Pennsylvanian petroleum consists of a mixture of hydrocarbons of the $C_nH_{2n+2}$ group in which $n$ may be anything from 1 to, say, 30, and the boiling point rises gradually from 0° C. for $C_4H_{10}$ to 280° C. for $C_{16}H_{34}$. Hexane ($C_6H_{14}$), which is practically **petrol,** has a boiling point of 69° C. and a density of 0·664.

The density of the oil on leaving the well is about 0·84 to 0·90, and the output for the whole world is in round figures 20,000,000 tons. This total will be observed to be a very small one when compared with the output of coal, which is about forty times as great. Unless therefore fresh supplies of oil are discovered it is no use to hope to replace solid by liquid fuel, although the higher calorific value

* Professor Hopkinson found the calorific value of a sample of Pratt's motor spirit (density = 0·715) to be 17,500 B.T.U. per pound on the lower value (i.e. assuming steam formed during combustion not to be condensed).

of the latter might make the change desirable. The sources of the supply of petroleum in 1907 were as follows—

| | | |
|---|---|---|
| Russia | 28·0 | per cent. |
| U.S.A. | 59·6 | ,, |
| Dutch Indies | 3·3 | ,, |
| Roumania | 2·9 | ,, |
| Galicia | 2·5 | ,, |
| India | 1·9 | ,, |
| Other countries | 1·8 | ,, |
| | 100·0 | ,, |

It will be seen that the first two countries produced nearly 90 per cent. of the whole.

**103. Coal Tar Products.**—Useful fuels are found among the **by-products of gas works** and, as will be explained later, some of them have been used successfully in internal combustion engines. Mr. O'Gorman has summarized the coal tar by-products as follows—

1. Ammoniacal liquor.
2. First light oils. } Crude naphtha.
3. Second light oils. } Crude naphtha.
4. Carbolic oils.
5. Anthracite oils.

2 and 3 can be broken up by distillation into :—

(*a*) Benzole.
(*b*) Solvent naphtha.
(*c*) Illuminating oils.

Benzole in turn distils to—

(i) "90 per cent. benzol" (so called because 90 per cent. distil out before 100° C.).
(ii) "50 per cent. benzol" (so called because 50 per cent. distil out before 100° C.).
(iii) Solvent naphtha.

**104. Ideal Conditions.**—The ideal fuel would be one which behaved uniformly in every part when subjected to an increasing temperature, one, for instance, which would begin to distil at a temperature quite close to that at which distillation ended. That the opposite condition to this is

an undesirable one will be understood by reflecting that if the temperature range of distillation be great there is a considerable chance that in the engine this fuel might be subjected to selective action such as would leave the heavier parts of the fuel as a deposit in the cylinder with consequent loss of horse-power and "gumming-up" of piston and valves.

The ideal fuel must be clean and easy to handle without danger. It should likewise be cheap. It is here that the **disadvantages of petrol** are so marked. It is both costly and dangerous, but is otherwise a good fuel because its range of distillation is small. The avoidance of danger requires the absence of low flash point, but on the other hand a low flash point fuel is good for starting the engine as there is then no need to heat the engine or carburettor first. If a low flash fuel is avoided the engine is less easy to start. On the whole, the ideal conditions are seen to be mutually conflicting.

Petroleum derivatives (i.e. everything from crude oil to petrol) are commonly so far lacking in uniformity of composition that the flashing point is governed by some small amount of lighter spirit which is present (even 1 or 2 per cent. will fix the flashing point) so that the measurement of this point does not tell one much as to the real nature of the bulk of the fuel. It does, it is true, tell one that the carburettor will need so much the less external heat to be applied, but of the closeness of the points over which the oil will distil it tells nothing. A medium heavy petrol (density 0·760) will distil completely between 60° C. and 150° C., a narrow range of but 90° C. Some oils, however, have a range of hundreds of degrees and would therefore be unsuitable for use in an internal combustion engine.*

The following practical test results have been obtained in the States on three exactly similar Maxwell cars driven by gasolene, kerosene and alcohol respectively over 249 miles of roads which were snow-covered in places to the depth of 10 inches—

* The best petrol has a specific gravity at 15°C of 0·715 to 0·730, and yields 63 per cent. of constitutents (by volume) at and below 100°C, and 90 per cent. at and below 120°C.

| Fuel. | Cost per Gallon. | Total consumption in Gallons. | Cost of Fuel per Car. | Cost per Mile. | Cost per Ton-mile | Miles per Gallon. |
|---|---|---|---|---|---|---|
| | $ | | $ | $ | $ | |
| Gasolene (or petrol) . . . | 0·20 | 24·75 | 4·95 | 0·19 | 0·0169 | 10·1 |
| Kerosene (or paraffin) . . | 0·13 | 33·75 | 4·39 | 0·17 | 0·0139 | 7·4 |
| Alcohol . . . | 0·37 | 40·75 | 15·07 | 0·60 | 0·448 | 6·1 |

**105.** These figures show that alcohol is not—in the United States at least—a cheap fuel and that kerosene is. The Fuels Committee of the Motor Union, however, appear to have considered **alcohol** as a possible alternative and rival to petrol, and their 1907 Report dealt largely with this possibility. The following extracts from the Report are given—

Most readers of this Report are familiar with the properties of petrol as a fuel, but they have very little idea of the great advantages of alcohol, having probably only heard of certain objections more or less imaginary, such as corrosion, and it has, therefore, been thought desirable to add the following summary of the properties of alcohol, comparing them with those of petrol :

(1) Safety, (2) thermal efficiency, (3) calorific value, (4) practical limit of compression, (5) complete combustion, (6) propagation of the flame, (7) smell, and (8) flexibility.

(1) *Safety.*—In the first place, in case of possible conflagration, alcohol can be extinguished by water, whereas petrol is only scattered under similar circumstances and the area of conflagration increased. In the second place, and even more important, the flash point is considerably higher, being 60° Cent. compared with petrol, which may be taken as anything down to 10° Cent. below freezing point. This enables the alcohol to be carried and stored with safety under conditions where petrol would not be permitted. This further very much reduces the cost of freight and insurance.

(2) *Thermal Efficiency.*—Owing to less air being required and a consequent reduction in the amount of inert gas, the thermal efficiency of alcohol is as high as 35 per cent., as against something below 20 per cent. in the case of petrol, and this greatly reduces the chances of overheating, besides also reducing the weight of cooling water, radiator, etc.

(3) *Calorific Value.*—The calorific value of absolute alcohol is 12,600 B.T.U., that of methyl alcohol with a specific gravity of 0·820 is 11,300, and alcohol with the addition of 20 per cent. of water shows a calorific value of 9,810 ; whereas that of petrol with a specific gravity of 0·722 ranges from 20,300 to 19,300 B.T.U.

(4) *Practical Limit of Compression.*—The practical limit of compression of alcohol is about **200 lb. per square inch.**; and its explosion pressure is therefore considerably higher than that of petrol, the practical limit of compression of which—in view of possible pre-ignition—is limited to 80 lb. per square inch.

(5) *Complete Combustion.*—With alcohol complete combustion is more easily attained, owing to the fact that it distils completely in its commercial form over a small range of temperature (80–100° Cent.), a very accurate degree of carburation thus being maintained. In the case of petrol the range of boiling point extends between 50° Cent. and 150° Cent.; such a large range of boiling points renders accurate carburation at all times more difficult, and makes the spirit what is commonly known as *stale* owing to the evaporation of the lighter fractions. Alcohol has not this disadvantage, the liquid being practically homogeneous throughout.

(6) *Propagation of Flame.*—There is less rapid propagation of the flame when alcohol is used, which gives a much more uniform pressure throughout the stroke than petrol.

(7) *Smell.*—With alcohol there is approximately no offensive smell in the exhaust, as compared with petrol.

(8) *Flexibility.*—Alcohol will explode when mixed with air over a wider range than petrol—4·13 per cent. alcohol vapour in air being combustible, the range in the case of petrol vapour being 2·5 per cent.; thus the engine will be much more flexible.

There are three points, however, on which it is popularly supposed that alcohol compares unfavourably with petrol. These are:

(9) Corrosive effect.
(10) Starting from cold.
(11) Vaporization.

(9) *Corrosive Effect.*—With regard to alcohol, any corrosive effect that may occur is probably due to impurities in the denaturing agent present in acetone and methyl alcohol, but these difficulties would be overcome if the carburation is such as to give complete combustion. Upon this point Dr. W. R. Ormandy writes to the committee as follows:

"My information with regard to the action of the effluent gases from motors running on alcohol was obtained from the engineer at the Gährungsversuchsanstallt at Berlin, who reported that engines running on pure alcohol, or even on pure alcohol with the German denaturant, gave no appreciable corrosion except on such parts of the motors as were so cold that condensation took place; thus the silencer was apt to corrode, more so the larger the percentage of water in the alcohol employed. As the average amount of water at present in German industrial alcohol is 10 per cent., this corrosion might become appreciable if the cooling of the cylinder walls was too effective. It has been proved, however, that the efficiency of alcohol engines is enormously increased by keeping the cylinder walls near the temperature of boiling water, and under these conditions no condensation and no corrosion obtained."

(10) *Starting from Cold.*—As for difficulty in starting from cold, it will be probable that alcohol as a fuel will almost always have a greater or less quantity of benzol mixed with it, in which case this

difficulty entirely disappears. Even without the addition of benzol there is little doubt that the question of starting from cold will be almost entirely overcome by the use of a suitable carburettor.

(11) *Vaporization.*—Alcohol requires $5\frac{1}{2}$ per cent. of its total heat of combustion to vaporize it, whereas, on the other hand, petrol vaporizes without any external assistance. With regard to the heat required to vaporize it, it is to be noted that, inasmuch as a large amount of the heat produced passes off in the exhaust, this is really available for the purpose of vaporization and does not represent any thermal loss.

*Other Means of Utilizing Alcohol.*—From the previous argument it will be seen that, in order to utilize alcohol in an internal combustion engine, certain modifications in the engine itself become necessary, but it is quite reasonable to expect that such alterations would be unnecessary if the proportion of tar benzol, acetylene, or other hydrocarbon containing a high percentage of carbon were mixed with the alcohol. Owing to this high percentage of carbon present, the chemical composition of the mixture will be brought more nearly to resemble that of the petroleum products. As to the most suitable relative proportions, experiment only will determine these, but such a fuel as is here suggested has the advantage of being a home production, as well as one that could be used without material alteration to the engine.

* * * * *

It has been stated in evidence that the average price at which alcohol can be produced in Germany amounts to 1*s.* a gallon, including the cost of denaturing and Government supervision. It is also a fact that in this country the actual cost of manufacturing alcohol amounts to $11\frac{1}{2}$*d.* a gallon (64 overproof, a strength common in industrial spirit)—see Report of Departmental Committee on Industrial Alcohol. This is produced from beet, potatoes, and molasses. Evidence has been given which tends to show that alcohol may also be produced from sawdust at a very low cost. The lowest figure it is possible to touch in this respect is 3*d.* per gallon when peat is used. Now, owing to the great strictness of the Excise authorities in England, the cost of denaturing and expenses of supervision bring the total cost of the alcohol up to about 2*s.* per gallon at the present time, and it is therefore evident that should the Government see their way to take a wider view of the question of alcohol as a fuel for internal combustion engines this price of 2*s.* a gallon could be very materially reduced. If this were done, the price could easily be brought to such a figure that it would be a very serious competitor with petrol in this respect alone. The Government that will recognize this, and will allow untaxed alcohol suitably denatured to be used for light, heat, or power, will be conferring an immense boon and benefiting a very large proportion of the population.

As regards the possible use of **benzol** the Committee remark:

What is commonly known as 90 per cent. benzol, can be utilized with perfect success in the engine of a motor car either alone or mixed with petrol, or mixed with alcohol. Owing to the high percentage of carbon which is found in benzol, and to the low

percentage of carbon in alcohol, it is evident that a mixture of these two liquids more nearly approaches the ordinary hydrocarbon liquid fuels to which we are accustomed in its chemical composition. Benzol will carburate air in the ordinary way when an ordinary petrol carburettor is used, but its specific gravity is very much higher than that of petrol, viz. 0·883, which may necessitate an adjustment of the float to prevent the benzol standing too low in the jet of the carburettor. Crude benzol inevitably contains a certain amount of foreign matter in combination with sulphur, which imparts to it an unpleasant smell in the liquid state. Owing to its comparatively low price, however, it might pay to have benzol still further treated after washing in order to remove these impurities, which could be done for the expense of about 1*d.* per gallon. At the present time benzol cannot be obtained in very large quantities, as the number of recovery plants in this country is not very large. As benzol is a home production, its use should be encouraged, and particularly at this present time when the difference between the prices of petrol and benzol is very small.

Mixtures of benzol and alcohol have been tried in a desultory manner on the Continent, but in this country nothing has been done upon an extensive scale. The possibilities for the successful use of such a mixture are very great, and both these fuels are capable of manufacture in this country in very large quantities. Although a mixture of benzol and alcohol is in its normal state quite nauseous, and would not require a further treatment such as the addition of wood naphtha, yet it is possible, at any rate, to partially separate these two liquids, the alcohol having an affinity for water.

**106. Sources of Petrol Supply.**—The advent of the motor car has been the main cause of the increase in the demand for petrol, which previously had been regarded as a waste product. Petrol is now one of the most valuable components of crude mineral oil. According to Mr. Duckham the following are the leading sources of the petroleum spirit (or petrol) imported to this country :—

| From | 1904. Per cent. | 1905. Per cent. | 1906. Per cent. |
|---|---|---|---|
| United States . . . . . . | 50 | 56 | 29·8 |
| Sumatra, East Indies, Borneo and Netherlands . . . . | 37 | 42 | 61·4 |
| Roumania . . . . . . . | 8·2 | — | 8·6 |
| Other Countries. . . . . | 4·8 | 2 | 0·2 |
| | 100·0 | 100·0 | 100·0 |

**107. Slow-speed Oil Engines.**—As an ordinary petrol engine

which is run on paraffin becomes thereby an "oil engine," it is necessary to bring in the variable factor, speed, in order to distinguish it from the older and heavier types of oil engine. The former runs at, say, 1,000 revolutions per minute, and the latter at only a few hundred—quite commonly 200, although there is not the same general constancy of speed in this class that there is in the other. Neither is essentially different from the other. If a petrol engine is imagined as greatly increased in size—say to a cylinder diameter of 14 in.—and all parts increased in proportion, the safe speed at which the engine will run will have to be reduced, because whilst the weight of moving parts goes up with the cube of the dimensions, sectional areas of stressed metal only increase as the square. It is not possible therefore to aim at high speeds without greatly increasing the cost of production. The building of such expensive engines is frankly put on one side and a cheap engine is built which will run at a very slow speed, slower even than in proportion to the increase in dimensions would naturally suggest. This enables cheaper materials to be used than are employed in the construction of petrol engines. The output in horse-power is reduced in proportion to the speed, but increased as the square of the cylinder dimension, provided that ports, etc., are designed of sufficient size to enable the working mixture to enter and leave the cylinder without undue obstruction.

As representative of this heavier class of engine the well-known **Campbell** oil engine is selected, and following on to it will be given a short description of the **Diesel** engine which works in a special way and is capable in proper hands of showing an exceedingly high thermal efficiency.

**108.** The **Campbell Oil Engine** is illustrated in Figs. 62 63, 64, 65, 66 and 67. Fig. 62 shows the engine to be somewhat similar in plan to a horizontal steam engine, and the engine parts are generally on that scale. The inlet valve *C* and exhaust valve *G* are shown in position. The latter is worked through a lever *H* and side rod *J* by an eccentric *K* driven from the crank-shaft *L* by spur gearing. When the speed exceeds the normal, a centrifugal governor

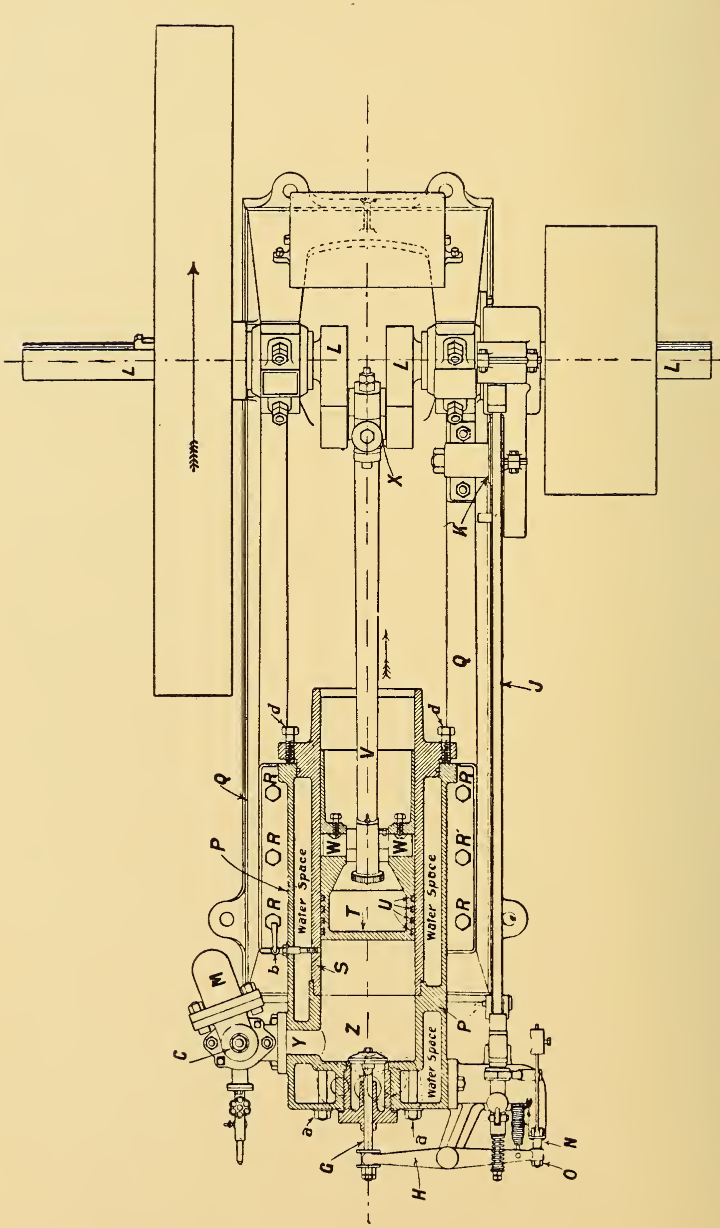



FIG. 62.—Sectional Plan through cylinder of Campbell Oil Engine.

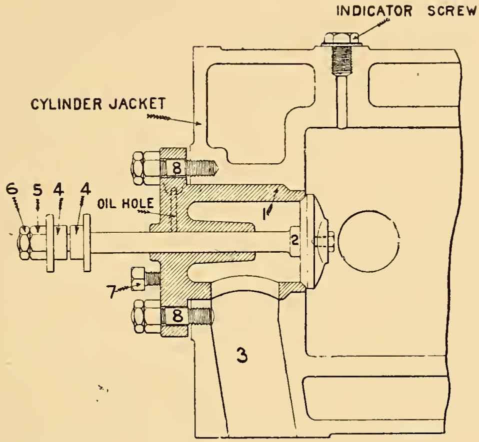



FIG. 63.—Section through exhaust valve and plug of Campbell Oil Engine.

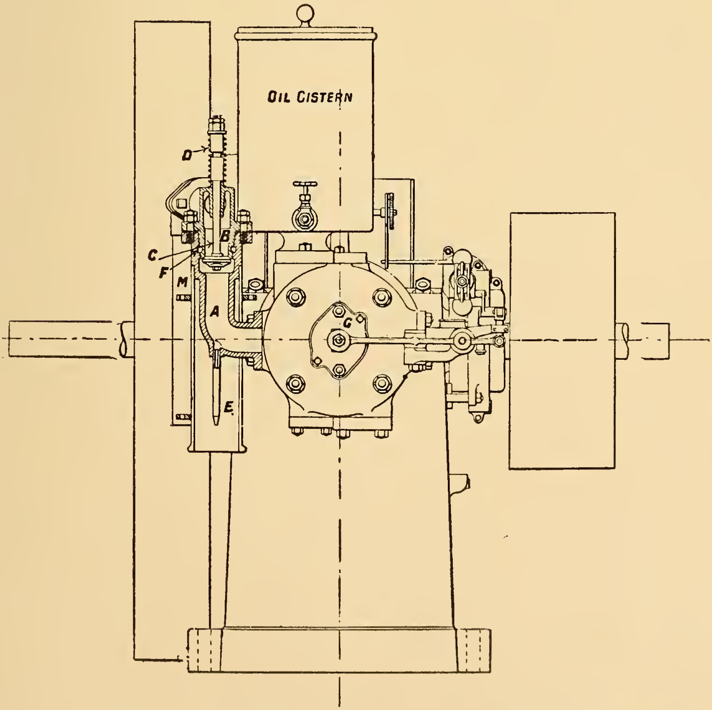



FIG. 64.—Campbell Oil Engine, illustrating operation of vaporizer

pushes down a steel piece $N$, which engages with a corresponding steel piece $O$ on the exhaust lever $H$, and prevents the exhaust valve $G$ from closing. When this valve is held open no partial vacuum can form in the cylinder during the charging stroke of the piston because there is free com-

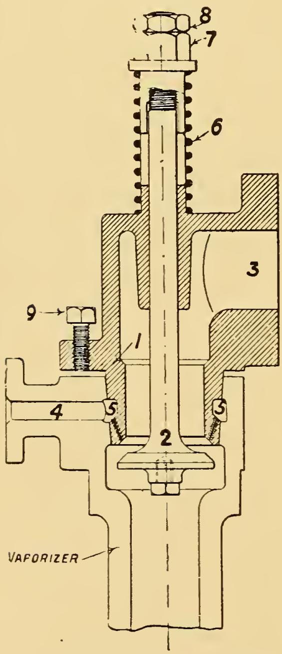



Fig. 65. — Section through inlet valve and plug of Campbell Oil Engine.

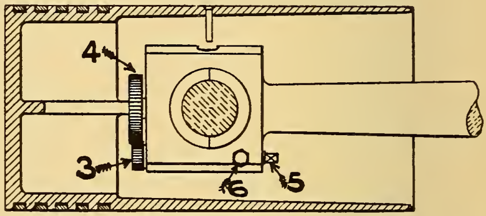



Fig. 66.—Campbell Oil Engine, piston detail.

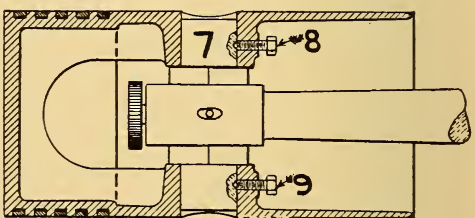



Fig. 67.—Campbell Oil Engine, piston detail.

munication with the atmosphere through the exhaust valve, and consequently no charge of oil and air can be drawn into the cylinder. The **vaporizer** for combining air and oil into an explosive mixture is shown in section in Fig. 64 and consists of a cast-iron chamber $A$ securely bolted to the cylinder and in direct communication with the combustion chamber. Into the top of this chamber the inlet valve plug $B$ is fitted and this plug contains the seat of the inlet valve $C$ (*see* Fig. 62). The inlet valve $C$ is

kept closed by a light spring $D$ and only opens during the charging stroke of the piston when a partial vacuum is formed in the cylinder. Oil is admitted through the annular space or groove $F$ and passes through small holes in the valve seat and into the vaporizer when the inlet valve leaves its seat. Air is admitted through the pipe $M$ and passes through the inside of valve plug $B$ carrying the oil with it. The ignition tube $E$ is screwed into a boss on the lower portion of the chamber $A$. The tube and the whole of the vaporizer is kept hot by an external lamp. During the charging stroke of the piston, a partial vacuum is formed in the cylinder and the charge of oil and air is drawn through the inlet valve, being sprayed during its passage against the heated sides of the chamber $A$ and thus vaporized. The mixture then passes into the cylinder, is compressed on the return stroke of the piston and then fired by the heat from the ignition tube. The timing of ignition is left to adjust itself, once the correct ignition has been found. Governing is, of course, by missing strokes and not by throttling.

**109.** The **Hornsby Type of Vaporizer** is also worth studying. This type of vaporizer is shown in Fig. 68, and to show how the vaporizer is fitted in place the diagram includes the cylinder also. When it is desired to start the engine a lamp is placed under the vaporizer chamber until the latter is at a sufficient temperature to ignite the oil which is pumped into it. This lamp is withdrawn once the engine is started as the heat of explosion is sufficient to keep the temperature up to the requisite point. The oil tank is under the engine and from it the oil is forced by a small pump into the vaporizer just at the moment when the piston is starting on its out-stroke and is drawing in the air necessary to combustion. The supply of oil is controlled by the governor in the following way. The oil passes through a valve-box with two valves, one of which leads to the vaporizer and the other leads to an overflow from which the oil can flow back to the tank. If the speed rises beyond the required point the governor opens this latter valve and the quantity of oil getting into the vaporizer is therefore reduced. On the return stroke of the piston the mixture

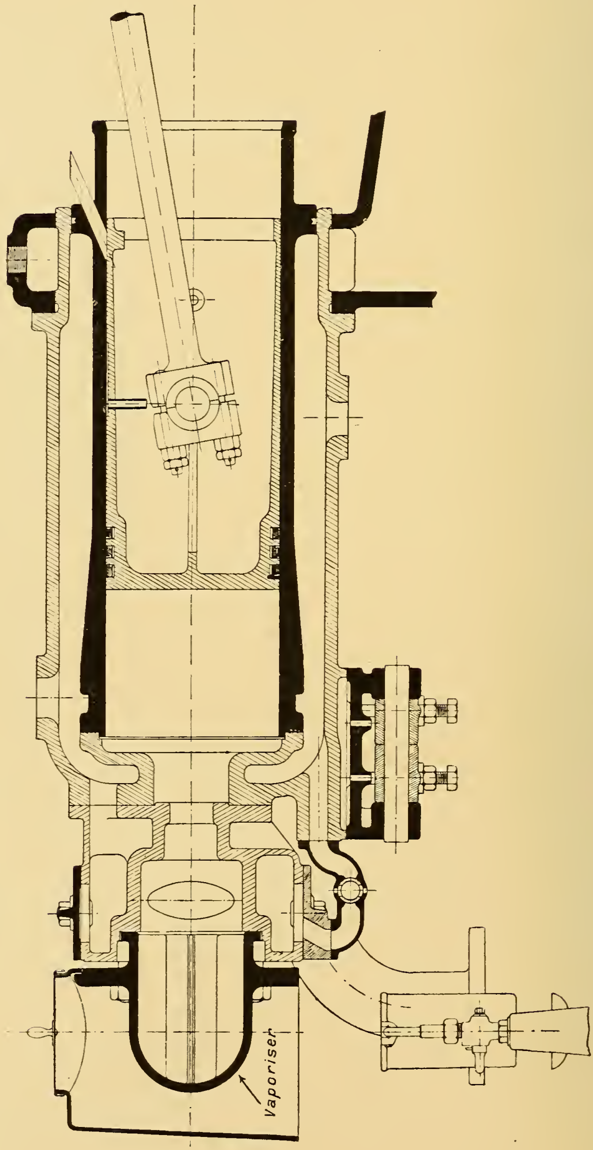



FIG. 68.—Section of cylinder and vaporizer—Hornsby Oil Engine.

is compressed and some of it forced back into the hot vaporizer where the temperature is so high that ignition

occurs and a working stroke is therefore made by the piston. The vaporizer chamber can, of course, be taken out and cleaned when desired. It is found, however, that even when working on quite heavy unpurified oils very occasional cleaning will suffice.

**110.** The **Diesel Engine** differs from the above type of slow-speed oil engine in that the oil is admitted gradually during **the explosion stroke** instead of during the suction stroke. It works on the Otto or four-cycle principle just as the Campbell engine does, but the details differ. Thus

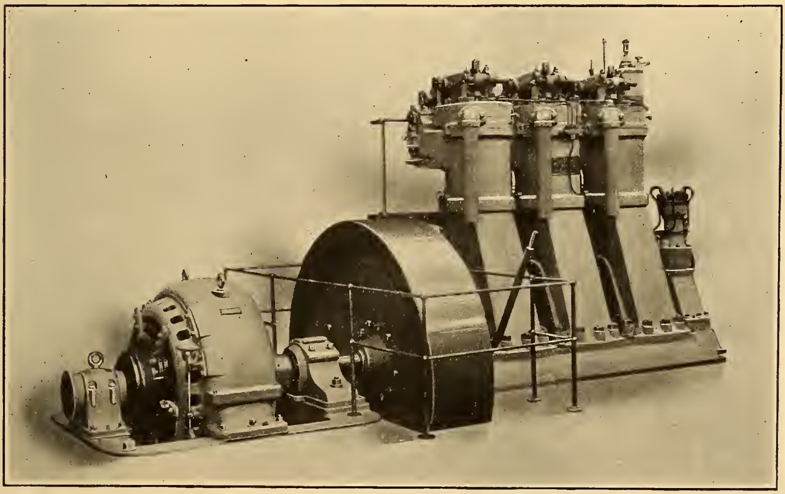


Fig. 69.—Three-cylinder "Mirrlees Diesel" Oil Engine coupled direct to 90 K.W. Generator. For Birkdale District Electric Supply Co.

in the Diesel engine (*see* Figs. 69, 70 and 71) air alone is taken in during the suction stroke. Air alone is compressed to 35 or 40 atmospheres and to a temperature of 1,000° F. (a dull red heat) or more. Then on the ensuing outward stroke oil is sprayed into the cylinder at such a rate as to produce during the first part of the stroke a nearly uniform pressure of about 500 lb. per square inch. At a given point in that stroke the fuel supply is cut off and expansion takes place. Then follows the usual scavenging stroke. The speed for a 240 h.p. engine is about 160 revolutions per minute. It is claimed that the fuel consumption

need not be more than 0·40 lb. of oil per b.h.p.-hour. If the oil have a calorific power of 15,000,000 ft.-lb. per pound, this is equivalent to the engine yielding 1,980,000 ft.-lb. (being one b.h.p.-hour) for every 0·40 ×15,000,000 ft.-lb. put into it, giving an efficiency of $\frac{1,980,000}{6,000,000}$ or almost exactly one-third or 33 per cent. This is a very high effi-

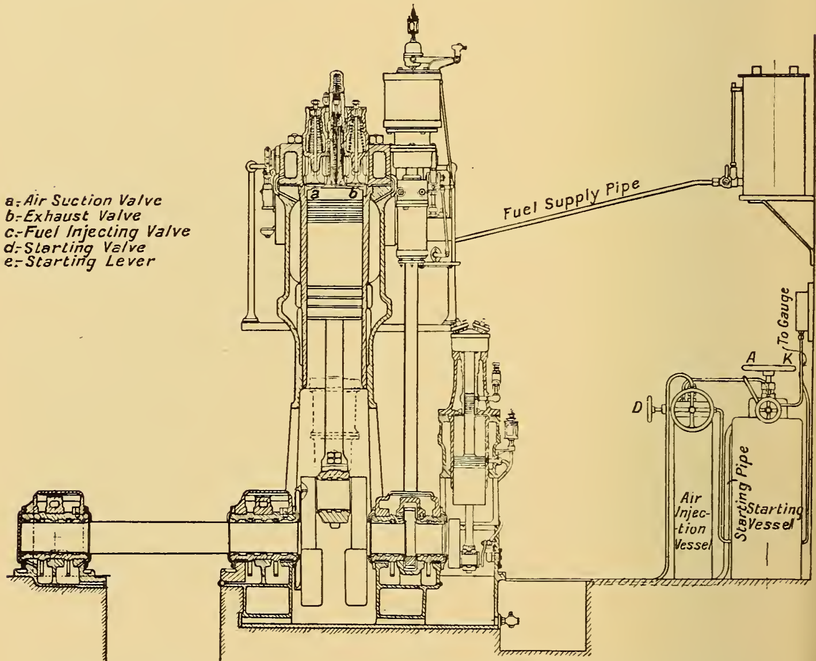



FIG. 70.—Sectional view of Diesel Oil Engine (Mirrlees, Watson and Co.). Note position of inlet valves. *See also* Fig. 71.

ciency, and the reason why it can be obtained is mainly on account of the high compression employed, viz. 35 or 40 as against 5 or 6 as commonly used on other engines. The Diesel engine cycle aims at compliance with the principle of taking in all its heat at constant pressure and the possible efficiency given by the formula $\eta = 1 - \left(\frac{1}{r}\right)^{\gamma-1}$ would when $r = 40$ be no less than 0·78 as compared with a cor-

responding figure of 0·48 when $r=5$. This is enough to show that so considerable a rise in compression ratio might

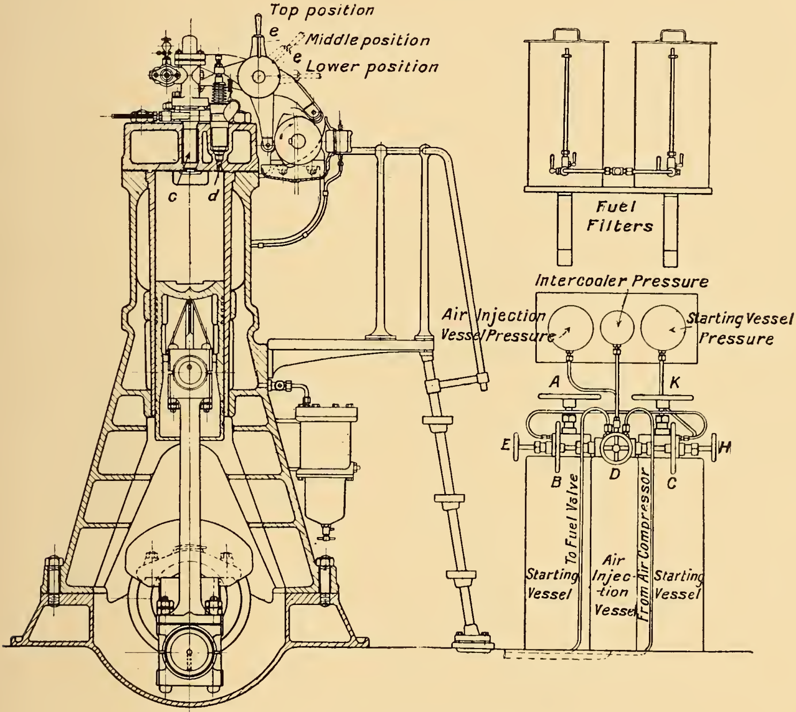



FIG. 71.—Side view of Engine shown in Fig. 70.

be expected to lead to a high thermal efficiency, and engine tests have confirmed this.

**111. The Thornycroft marine engine** can be operated with either petrol or paraffin. It is, of course, easier to work a marine engine on paraffin than a land one, as in the former the starting torque required is very slight and the speed at which the engine runs is much more even. There are, in short, no hills to climb.

The Thornycroft engine is illustrated in Figs. 72 and 73, and the following description will help to elucidate them.

In the first place it will be noticed that the engine is essentially a marine one, the bearing arms being cast on

the bottom half of the bed-plate, and large doors being fitted in the upper half to enable adjustments to be made to the bearings, etc. It will also be noticed that the engine is substantially built and suitable for heavy continuous running at full power.

The cylinders *M* are cast in pairs with large water-jackets

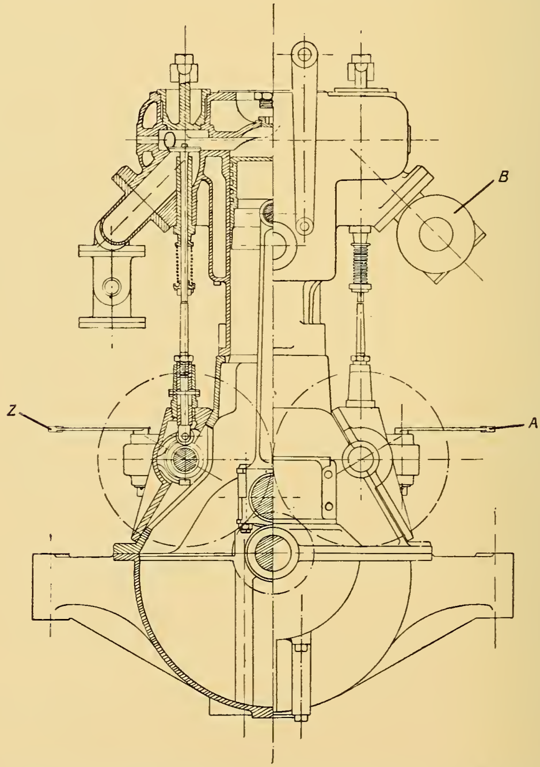



Fig. 72. General Arrangement of Thornycroft 6″ × 8″ Marine Petrol or Paraffin Engine—End view.

*N* surrounding them ; these water-jackets extend sufficiently far down to enable the working parts of the cylinders to be completely covered. *O* is the piston fitted with five piston rings ; *P* the connecting rod working on the gudgeon pin *Q* fitted with a solid bush. *R* is the crankshaft, and it will be noticed that the cranks are at 180 degrees with each

other. The main bearings are shown at $S$ and are of considerable length. The bottom ends $T$ of the connecting

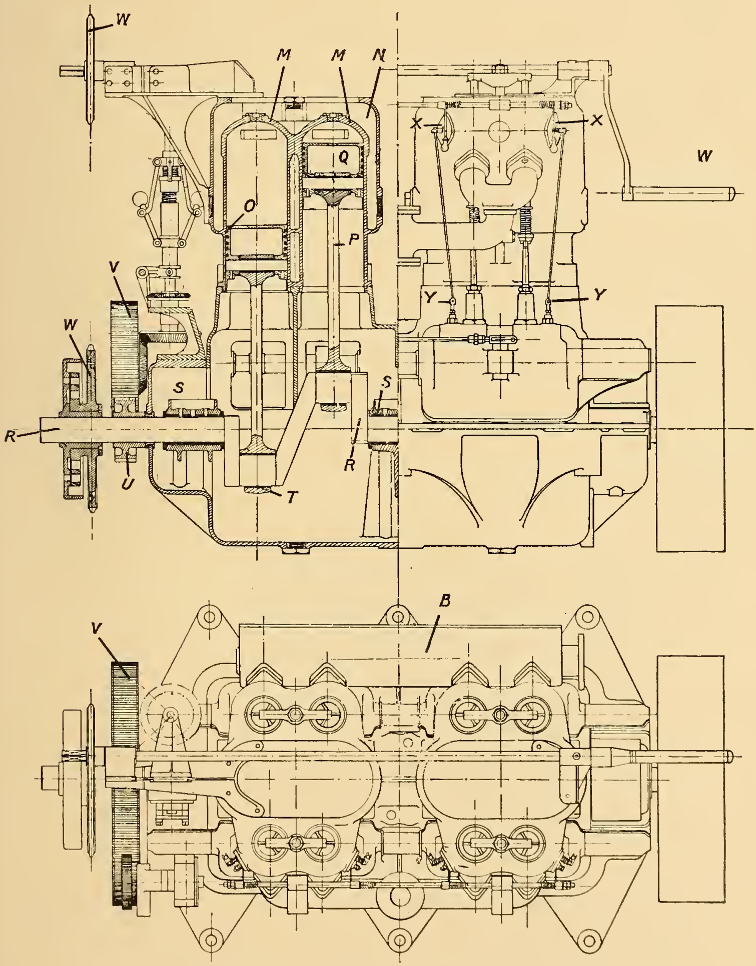



FIG. 73.—General arrangement of Thornycroft 6″ × 8″ Marine Petrol or Paraffin Engine—Side view.

rods are adjustable, and it will be noticed that to assist lubrication the cap and bottom half brasses are left slightly narrower than the top half. The pinion $U$ on the crankshaft drives two fibre wheels $V$ connected to the half-speed

shafts. The free-wheel starting arrangement is shown at *W*, together with the handle and chain wheels. The sparking plugs are shown at *XX* and are of the positive make-and-break type worked by tappets *YY*. Advance sparking gear is worked by the lever *Z*, and half compression for starting by the lever shown. The exhaust collecting-branch is water-cooled.

The makers claim that this engine is exceedingly simple to work even when very little practical knowledge is available; the motor canoes *Spider* and *Sandfly* supplied to Southern Nigeria have for some time been running under the care of native drivers, and they proceed long distances up the Cross River without white supervision. It is stated that the *Spider* had at the end of November, 1907, done 10,000 miles, and the *Sandfly* somewhat less; the *Spider* is a screw-in-tunnel boat, and the *Sandfly* a stern-wheel boat.

**112. The Petrol Engine.**—The principle of working in a

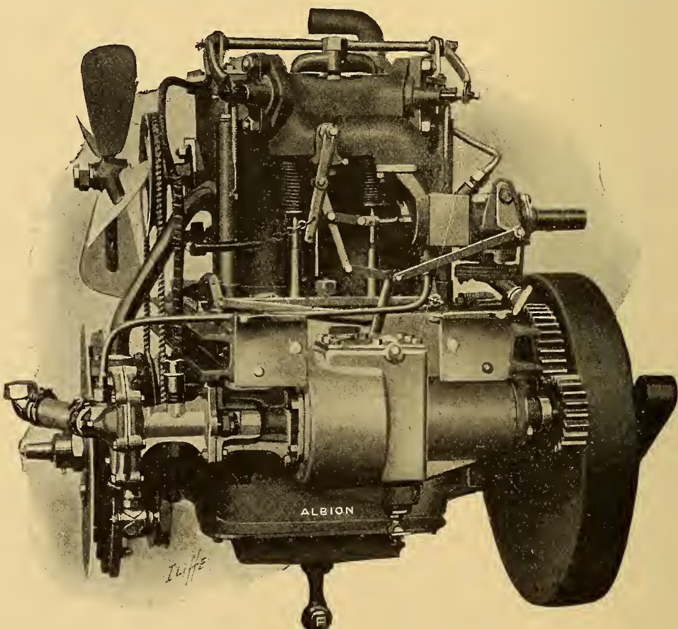



FIG. 74.—16 H.P. Two-cylinder Albion Engine.

petrol engine is **just the same as that of a gas or oil engine**—so much so that petrol engines have not infrequently been coupled up to suction producers and run as gas engines. Although this is so it must be borne in mind that owing to differences in the nature of the working fluid the proportions of the engines require to be designed separately for each method of working. Thus the ports of a petrol engine will be too small for efficient working as a gas engine. In a petrol engine the working fluid is a mixture of air with about 2 per cent., by weight, of petrol vapour. This mixture is formed by admitting both air and petrol to a device called a carburettor (about which more will be said presently). From the carburettor the mixture passes to the engine — most often through a throttle valve of the butterfly wing variety. The proportions of air and petrol are

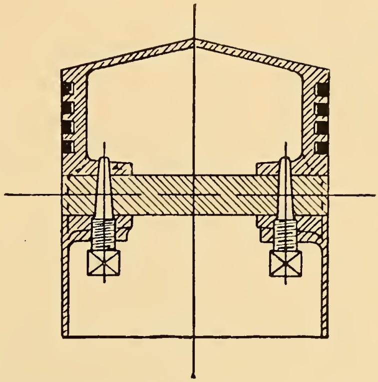


FIG. 75. — Typical piston of petrol engine showing covering of head and method of fastening the gudgeon pin upon which the connecting rod works.

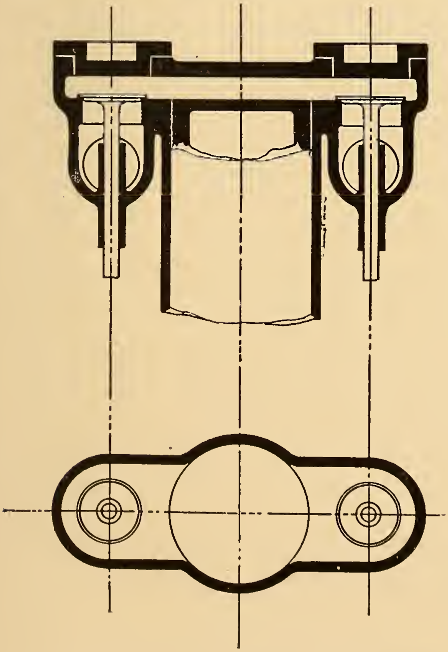


FIG. 76.—Section through cylinder having inlet and exhaust valves on opposite sides —as for instance in Albion Engine.

adjusted by having variable inlets for the air and controlling them by hand or by a governor. Illustrations are shown of petrol engines of the Albion, Lanchester and other types. Both are well known and deservedly popular.

One, two, three, four, six or eight cylinders may be used to make up one engine. The cheaper cars usually have

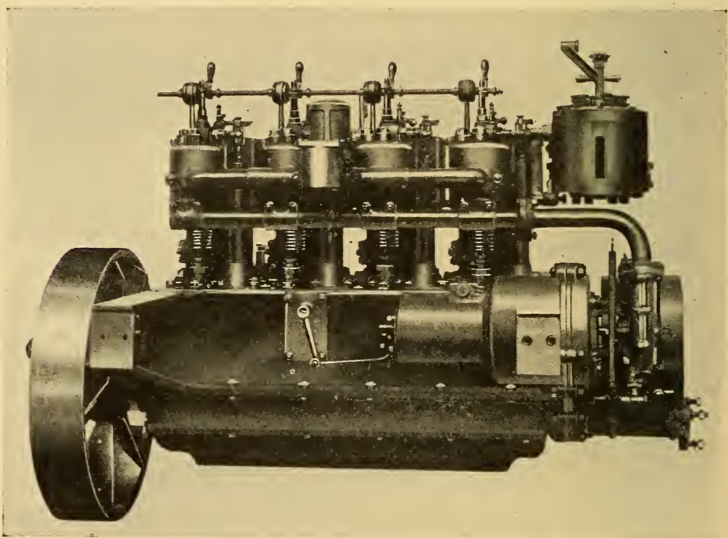


Fig. 77.—24 H.P. Four-cylinder Albion Engine.

one or two, the four-cylinder car represents the medium, and six cylinders are most commonly fitted to the very best cars. Eight cylinders have only been tried experimentally so far. Marine engines may have any number of cylinders. The more cylinders an engine has the more uniform is the turning moment and the lower the speed at which the engine can be run without stopping. This is an important point, and it is usually discussed under the title of " flexibility." A common speed for full load working is 1,000 revolutions per minute, and it is often convenient to be able to run at much lower speeds. Throttling the mixture has this effect and with a six-cylinder engine there should be no difficulty in getting down to 150 revolutions per minute. If one attempted to do this with a

one or two-cylinder car the result would be to stop the engine. To get lower speeds, therefore, with small engines one has

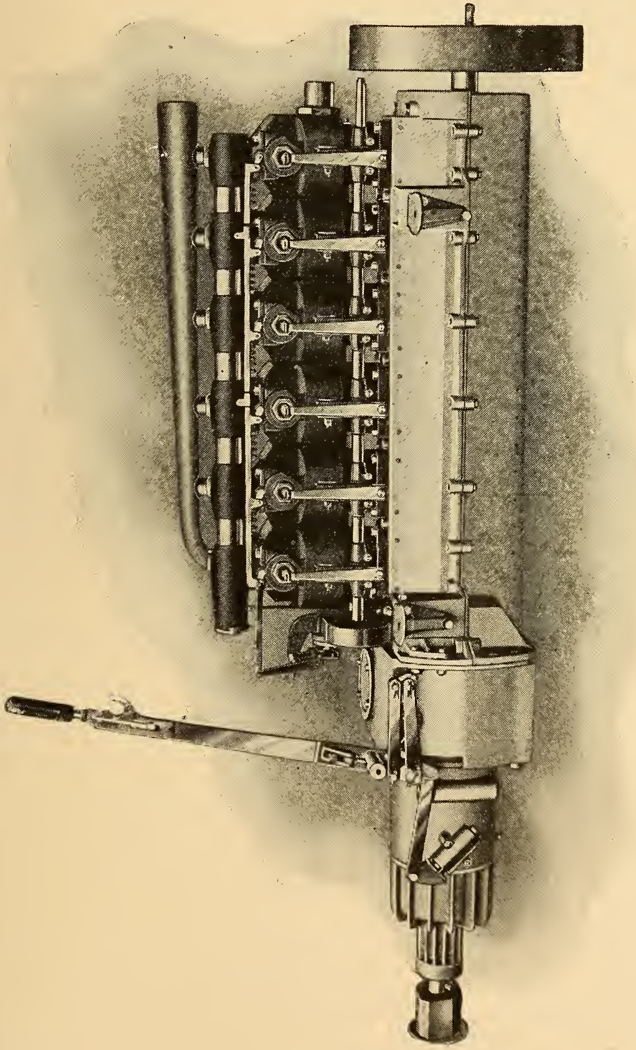


Fig. 78.—28 H.P. Six-cylinder Lanchester Engine.

to "change speed" as it is called. This brings us to the consideration of the mechanism by which the power of a petrol engine is transmitted to the road wheels of a car. The reader

will best understand this by forming a picture in his mind thus (*see* Fig. 80) :—The engine is fitted to the car so that the crankshaft points in the direction of motion of the car ; this shaft is continued from the front of the car to the back and its continuation is called the propeller shaft, owing to its being similarly placed to the propeller shaft of a screw steamship and actually being the propeller shaft when used in a

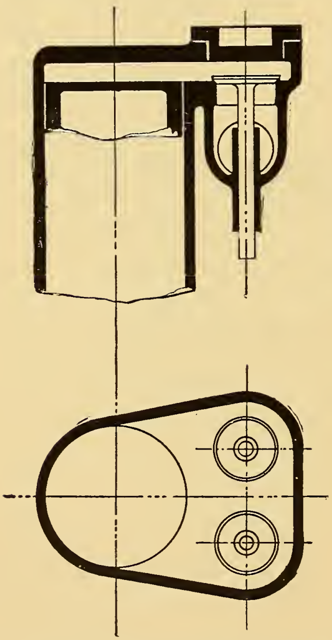


Fig. 79.—Section through cylinder having both valves on same side.

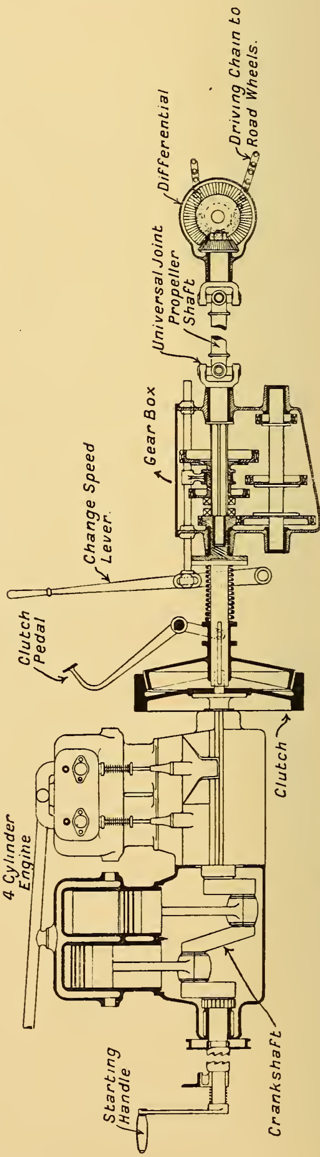



Fig. 80.—Typical arrangement of Petrol Engine for driving a motor vehicle.

marine motor. This shaft transmits power by bevel or

chain gearing to the rear shaft of the car, which of course is at right angles to it. In order to be able to alter the velocity-ratio between the engine and rear shaft a gear box is fitted to the end of the propeller shaft which acts like the back-gear of a lathe, that is to say by sliding gear wheels in and out of action, by means of a lever worked by hand, the velocity-ratio can be altered at will.

Readers who are familiar with motor cars and their engines will notice that the engine has been spoken of as being on the front of the car. This is by far the most common practice, although cars are built, such as the 10 h.p. Adams, in which the engine is placed under the back part of the body of the car. It is simpler, however, to the novice to picture the most common form of disposition of parts and to inquire into the others afterwards.

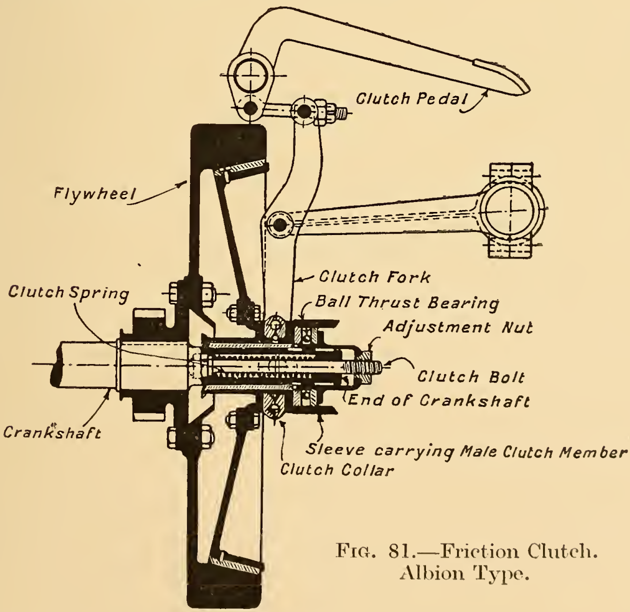



Fig. 81.—Friction Clutch. Albion Type.

**113.** Reverting to **the typical car** we have been thinking about, with the engine well forward under its "bonnet"—it is now desired, let us say, to start it. To do this it is necessary to give it a few turns by hand and for there to be

no load in the engine whilst doing this. To remove *all* load it is necessary to disconnect the engine temporarily from its propeller shaft and from the gear-box, differential, etc., which the propeller shaft drives. To do this **a cone clutch** is introduced between the engine shaft and the propeller-shaft. Fig. 81 is an illustration of such a clutch. The clutch consists of two conical parts, the inner one having a leather facing, which are pressed together by a spring and separated by depressing the clutch pedal. When the two parts are in frictional contact the engine drives right through, but when the clutch pedal is pressed down, only the left half rotates. It is in this latter position when the engine is started. To get the car into motion the foot is gently raised, and the two parts of the clutch come into contact and rotate as one piece. There are other varieties of clutch, but this is the commonest. The clutch is often made a good deal heavier than would otherwise be necessary, in order to provide a "flywheel effect," which is of great use to the engine, especially when there are only one or two cylinders and there are therefore several strokes in which no explosion occurs. The **gear-box** takes the most different forms according to the type of engine selected. From the gear-box a short length of shaft runs to the bevel gear in the "**differential**" as shown in Fig. 80. This gearing is similar to that used on old-fashioned tricycles and enables the two road wheels to adjust their relative speed when turning corners. The actual transmission to the road wheels is either by—

(*a*) Chain drive,
(*b*) Live axle drive, or
(*c*) Pinion drive.

In the first named alternative, chains are used much as in a bicycle. The disadvantages are noise and stretching of the chains under the sudden and heavy loading applied. The live axle drive is now becoming the more popular and is about 3 per cent. more efficient than the chain in respect of power transmission. The pinion drive is used for heavy vehicles for which the other two types would be unsuitable.

The **mechanical efficiency** of the transmission from engine cylinder to road wheels is variously stated as anything

from 60 to 80 per cent. The lower figure is the nearer one, and probably 65 per cent. is as much as can be expected. The following table shows generally the way in which the losses are incurred.

| | | |
|---|---|---|
| Power available at road wheels . | 65 | per cent. |
| Lost in gear-box . . . . | 15 | ,, |
| Lost in engine friction . . . | 10 | ,, |
| Lost in differential . . . | 7 | ,, |
| Lost in clutch . . . . | 1 | ,, |
| Lost in drive . . . . | 2 | ,, |
| I.H.P. developed in cylinder. . | 100 | ,, |

The author has travelled on cars driven by one, two, four, six and eight cylinders at speeds varying between one and sixty miles per hour, and on cars fitted with each of the alternative devices above mentioned. It is wonderful that such generally similar and generally satisfactory results are achieved in each case, but the difficulties which have yet to be surmounted are the ensuring of long life of the engine and other parts of the car, the attainment of high thermal economy of the engine and, what goes with it, *the complete burning of the petrol* used so that no carbon monoxide or hydrogen can be found in the exhaust.

The above description and illustrations of important parts should render the reader capable of following the more detailed treatment of the vital part of the mechanism which now follows.*

**114. Carburettors.**—The function of a carburettor is to intermingle the petrol or other fuel with the air so that an explosive mixture is formed which can be admitted forthwith to the cylinder. It is possible to allow only a portion of the air to pass through the carburettor and then to add additional air to the mixture so as to bring it to the required proportional composition. Or, on the other hand, the whole of the air may be passed through the carburettor. Both methods are commonly in use, often combined on one engine as will presently be seen. When petrol is used the "inter-

* Readers desiring further information as to the construction and working of motor cars should read Mr. Strickland's "*Manual of Petrol Motors and Motor Cars.*

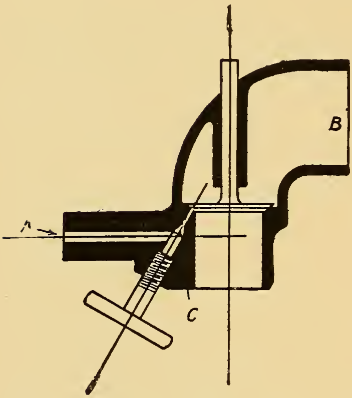



FIG. 82.—*A*. Petrol inlet to Controller. *B*. Inlet pipe to Engine. *C*. Needle valve to regular flow of petrol.

mingling" is largely between the vapour of petrol and air. Some petrol however comes over in the form of a liquid spray, and air carrying such a spray (or even coal-dust for example) is quite easily explosive.

The "intermingling" is caused in one of two ways: (1) by the **jet method,** or (2) by the **surface evaporation method.** In the former the jet may be of either of the varieties shown in Figs. 82 and 83. In the former the air sucked through *B*, on the opening of the valve, causes petrol to rush up the pipe *A*, past the screw-adjusted inlet opening to a small hole on the conical seating of the valve. The lift of the valve therefore not only admits air but uncovers the small petrol hole up which a jet of petrol at once squirts. Then the petrol, having so low a vaporizing point, at once turns into vapour and forms with the air an explosive mixture. The screw adjustment—or needle valve as it is called—allows the richness of the mixture to be adjusted. Fig. 83 shows a better and more familiar way of doing the same thing. Air is sucked in past the jet *F* and out by the opening *K*. In rushing past the jet it sucks up a petrol spray which evaporates as it gets mixed with the air. On the

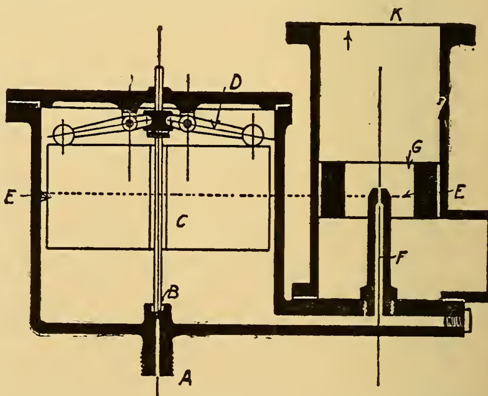



FIG. 83. — Jet Carburettor and Float Chamber. *A*. Petrol inlet. *B*. needle valve. *C*. Float which closes the needle valve *B* through the levers *D* when the petrol reaches the level *EE*. *F*. Petrol jet. *G*. Air nozzle. *K*. Inlet pipe to engine.

left of the figure is seen what is called a float chamber for keeping the petrol level constant. It operates much as does a ball and cock feed to a water cistern. *B* is a needle valve which gets pushed down on to its seating by the levers *D* when the float *C* rises to the top. This stops more petrol coming in until the petrol-level sinks so much as to let the float down till the levers open the needle valve again when more petrol flows in. The weight of the float is so adjusted that the petrol-level is kept at just the right height. It is the custom to have the petrol standing just below the top of the jet, but it works even if standing much below the top of the jet. Evidence of this has been

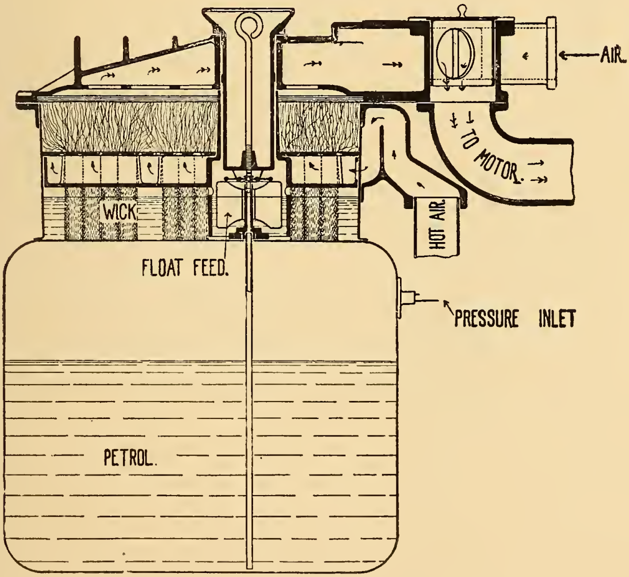



Fig. 84.—Lanchester Surface Carburettor.

afforded to the author by noticing instances in which the float chamber has been sucked quite dry during the running of the engine when the pipe *A* has got choked up in some way. The principle of the working of the jet will be gone into later. An efficient type of a surface carburettor is shown in Fig. 84, which illustrates the carburettor used on the Lanchester engine. The principle of its working is obvious from the

diagram. The air passing over a large petrol-soaked surface takes up petrol vapour. All the carburettors described work best with a certain velocity of flow of the air. When, however, the engine runs very fast or very slow the air velocity changes accordingly so that the carburettor sometimes gets more air, and sometimes less, than it wants. If the flow of

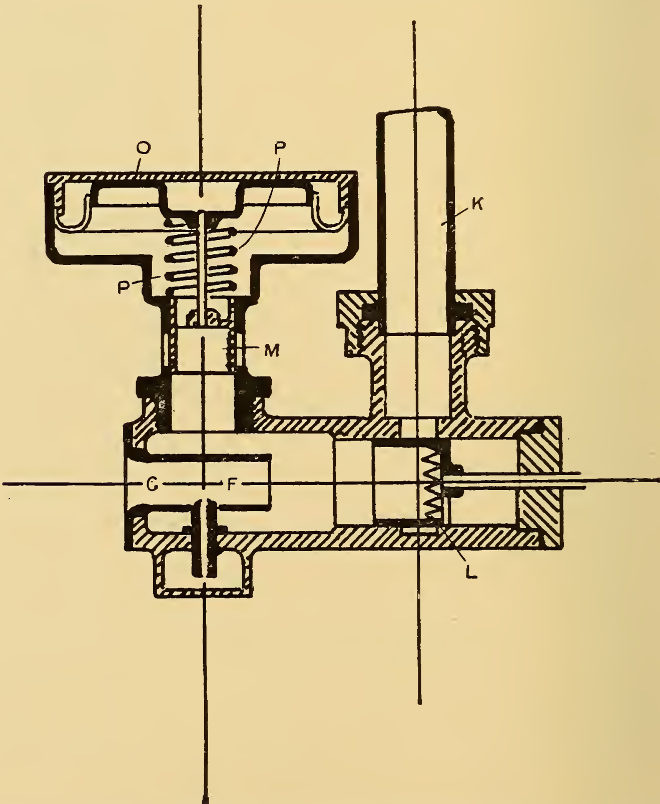



FIG. 85.—Krebs Carburettor, in which the opening to the extra air supply is controlled by the suction of the engine. It works excellently.

air be increased it is found that too much petrol is taken up, so it is customary to arrange for only part of the air to pass the jet (say) and for the rest to be added to the mixture without passing the jet at all. It is best to arrange for this adjustment to be made automatically and Figs. 85

and 86 show how this can be done. The former shows the Krebs automatic carburettor. When, owing to increase of piston speed, the suction on the air increases, the leather diaphragm *O* is sucked down against the weak spring *P* and opens a valve at *M* so that air can flow in and mingle with the air which has entered at *G* and has passed the jet *F*. *L* is the throttle valve controlling the quantity of the mixture which is allowed to pass to the cylinder by the pipe *K*. Fig. 86 shows another way of doing the same thing. As the suction increases the extra air comes in through the valve *M* and joins at *K* the part which has come

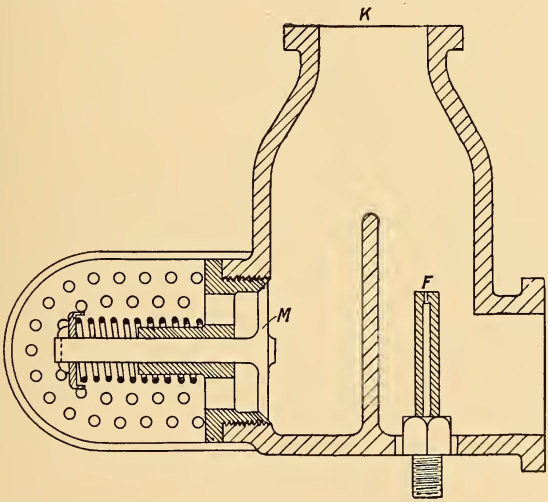



Fig. 86.—Automatic Carburettor working in a generally similar way to the Krebs. The opening of the valve *M* depends on the suction.

in over the jet *F*. There are many other ways, easily devised, of applying the same principle.

Owing to the heat absorbed by the evaporation of the petrol it is usual to warm the entering air slightly. This is done by putting the air inlet pipe nozzle close up to one of the exhaust pipes so that the air in rushing past the hot pipe gets warmed slightly. Of course the fact that the whole of the carburettor is under the warm engine bonnet helps to keep the temperature from falling too low.

When paraffin is used as a fuel much more heat is neces-

sary, as paraffin does not vaporize nearly so easily as petrol. The general form of float and jet is, however, the same.

**115.** The carburettor is usually controlled by **the governor** actuating a throttle, although sometimes it is the lift of the exhaust valves that is regulated. The former is the more common method, and as an instance of it the following description of the Murray governor is given. It is an instance of a governor which does as much as any governor

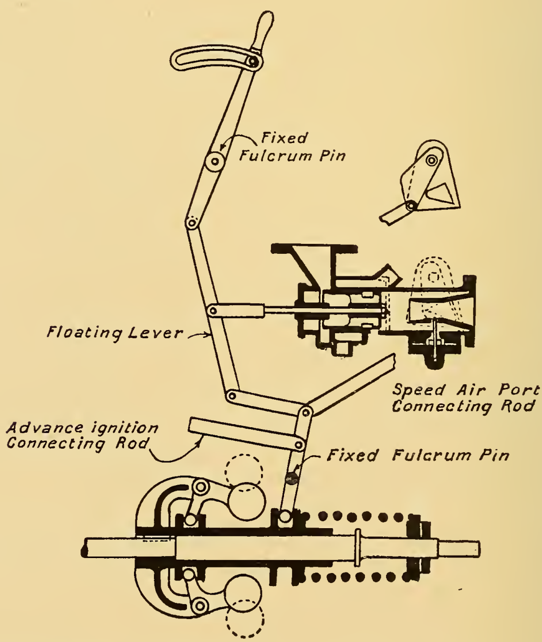



FIG. 87.—Arrangement of Murray Governor and Automatic Carburettor.

fitted to any car and far more than most. It controls the throttle, the ignition and the extra air port. By leaving out one or other of these any other type of governor is sufficiently well represented. Mr. Murray thus describes it : "The rotary portion of the governor is of the usual centrifugal type, but instead of being designed to act at one given speed the sleeve actuated by the centrifugal pull of the balls against the spring begins to move off its lower stop at an engine speed of 180 revolutions per minute and

does not reach the top limit of its travel until the engine attains a speed of 950 revolutions per minute. The travel of the collar along the shaft is much greater than is usual in centrifugal governors. The connecting link from the governor sleeve to the throttle valve is of a length variable by the control lever. In other words, the relative position of the throttle valve to the governor sleeve can be varied by the driver. This control lever therefore fixes the speed or point of travel of the governor sleeve at which cut-off is to take place, and thus sets the engine instantly to run

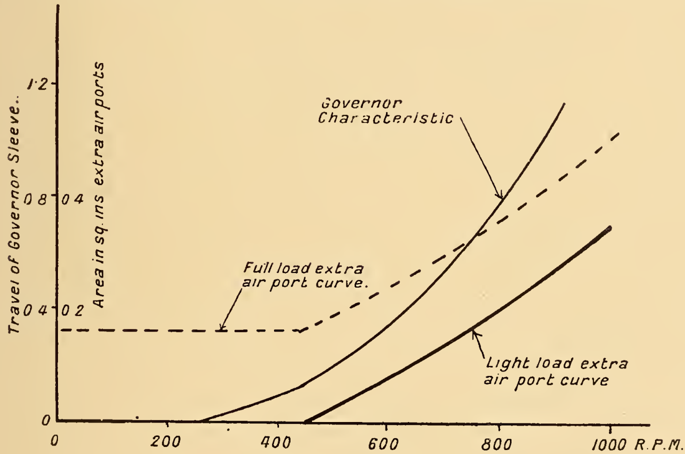



FIG. 88.—Characteristic Curves for Murray Governor.

at any desired speed within the given range. The throttle valve is of the piston type, so that it can overshoot its cut-off position in either direction by an amount equal to the total travel that the governor sleeve can give it. In its lowest position the control lever sets the throttle valve just open when the motor is at rest, therefore the moment the motor is started up and reaches a speed of 180 revolutions per minute the throttle valve commences to close and the engine will run about 200 revolutions per minute light. If set about one-third up, the engine will attain a speed of about 600 revolutions per minute, when the governor will once more close the throttle valve, for, as will be seen by a reference to the governor characteristic in Fig. 88 this speed corresponds

to about one-third of the travel of the governor sleeve. If the control lever is set right up to its control limit, the engine will rise to 950 revolutions per minute before the governor closes the throttle. About one-eighth of the total travel of the governor sleeve is all that is required to give practically full admission, so that when set at any given speed for light load a comparatively small drop in speed will suffice to open the throttle full up. Obviously, therefore, at whatever speed the driver sets the control lever, the governor will hold the car at practically a constant speed on the level, uphill and downhill, without any intervention on the part of the driver ; so long, of course, as the gradient is not too steep for the engine to tackle, on the gear on which it is running, or the down grade so steep that it is sufficient to drive the engine above its normal speed even with all petrol cut off. The throttle valve is attached to the centre of a double lever, one end of which is controlled by the governor, the other by the control lever under the direction of the driver. Setting the control lever at the lowest point, the governor being at rest, the throttle is just open, consequently a very small speed of the engine is sufficient to close the throttle throughout the range of the control lever. At whatever speed it is set the engine will follow up to this speed, and the governor will hold it there."

The author has had experience of the manner of working of this governor both by actual driving and by watching its action, and he has formed a high opinion of it. It performs its manifold functions better than the average man, but not, of course, so well as the specially skilled driver. As will be understood on reference to the diagrams shown—the governor is aimed to deal with the *average* case.

The type of carburettor generally used with this governor is shown in larger size in Fig. 89. In the annular chamber round the *vena contracta* (as it is called), at the right side of the illustration, there is an air port—called the speed air inlet—covered by a sliding shutter (*see* dotted lines) which is connected to the mechanism of the governor as already illustrated. The shape of the port is so designed as to give the area of

opening shown to be necessary by the curves experimentally obtained (*see* Fig. 88). Another port—called the load air inlet—is opened proportionately to the opening of the throttle valve so that the mixture may be kept constant as

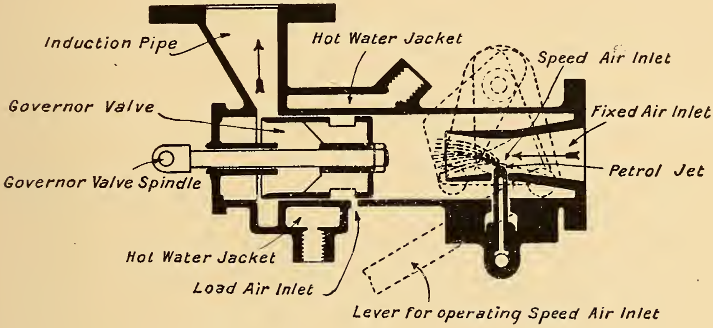



FIG. 89.—Albion Carburettor.

far as possible and free from any interference on the part of the driver.

**116.** A **carburettor** which has no additional air inlet and relies upon keeping the correct mixture by **manipulating the jet** is the White & Poppe, which is fitted on the Maudslay and other engines. The principle is to keep the flow of air past the jet constant as regards its *velocity*, and to arrange for the throttle opening to be proportional to the size of the opening through which the petrol passes up the jet. This is done by placing the jet in the centre of a chamber of circular cross-section across which the stream of air passes. This chamber is encased in a metal sleeve and the whole has a circular air-way drilled through both sides. Then as the jet chamber is caused to rotate slightly the air passage through is restricted and this restriction is made to affect the jet also owing to the hole of the jet being drilled a little eccentrically and a cap fitting on to the jet being similarly drilled and so fixed that as the jet and chamber rotate through an angle the effective jet aperture is decreased at the same time as the area through the air passage. That these two openings increase and decrease in the same ratio is ensured by the fact that in each case it is a circle

sliding over a circle and that both are fully opened and fully closed together. It is stated that with this carburettor as much as nine to ten miles per gallon have been accomplished on a car which otherwise would probably have only run seven miles. In the 1908 Commercial Vehicle Trials of the Royal Automobile Club a Maudslay vehicle fitted with this carburettor ran no less that 9·05 miles per gallon, a performance greatly in advance of that of any other vehicle.*

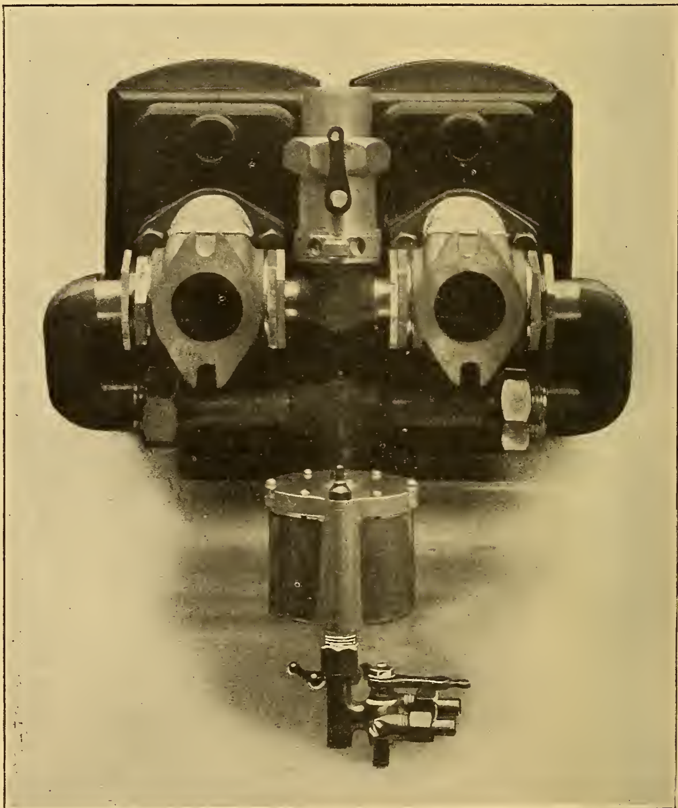


FIG. 90.—Cottrell Carburettor as applied to Two-cylinder Engine.

**117. The Cottrell Paraffin Carburettor.**—This carburettor is illustrated in Figs. 90 and 91. It is also shown diagram-

* *See also* Chapter IX.

matically in Fig. 92. In this diagram $C$ is the pipe which receives the air and paraffin spray coming over from the jet in the carburettor marked $H$ in Fig. 92. When the

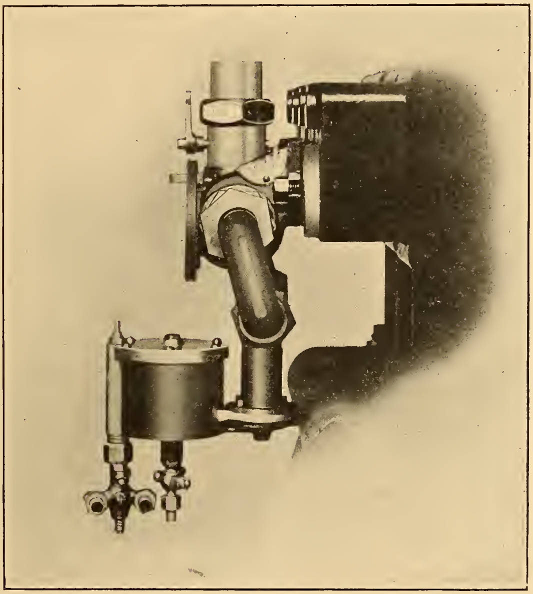


FIG. 91.—Side view of Cottrell Carburettor.

fuel gets to the branch pipe it divides right and left to either end of the vaporizer, $M$. $M$ is shown in section at the lower right-hand side of the figure. It consists of a corrugated pipe which is surrounded by hot exhaust gases and conveys in its interior the air and paraffin mixture. This corrugated pipe has to be kept hot. Paraffin will not work the engine until it is hot. In starting from the cold, provision is therefore made for working on petrol for two or three minutes and then, the pipes $M$ having got hot, the paraffin is turned on. At $B$ there passes a mixture of heated air and paraffin vapour. The air is much less in proportion than would work the engine, so at $F$ an adjustable inlet is fixed to admit more air until the mixture is of the usual proportions. The idea of not letting all this air in before the vaporizer is reached is that with a less propor-

tion of air the paraffin particles get more effectively heated and the velocity of passage through the vaporizing tube is slower. The whole process is therefore quite simple.

To recapitulate—a little air with a heavy proportion of paraffin spray is passed through a very hot corrugated tube

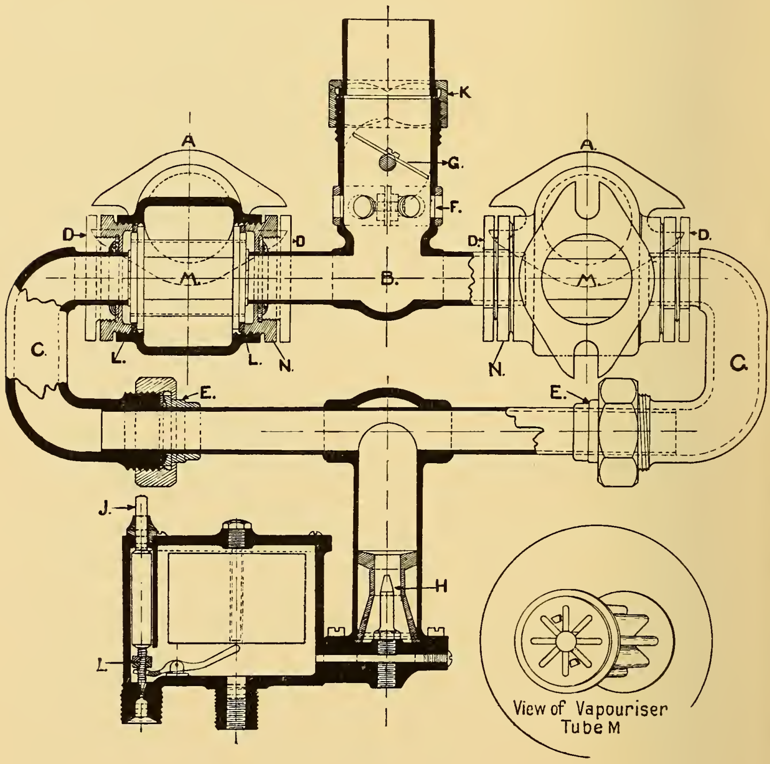



Fig. 92.—Cottrell Paraffin Carburettor. Shown diagrammatically.

and vaporized. Then it is diluted with a lot of cold air to the correct proportion and passed into the inlet port of the cylinder. These carburettors have been known to work successfully on motor cars for many thousand miles. A variation of it has been tried whereby air only was passed

through the star tubes *M*, then through a lagged pipe to the other side of the cylinder and thence through an ordinary carburettor such as is designed for use with petrol. This alternative arrangement also worked well, but hardly seems so effective as the unmodified type of Cottrell carburettor. To begin with, the paraffin draws all its heat from the air which sweeps it along, instead of by actual rushing contact with the hot vaporizer tubes. Further, the mixture enters the cylinder much sooner after its creation than in the other arrangement, and it is evidently advantageous to allow time for the paraffin vapour and air to mix intimately with each other. The patentees state that the compression pressure for working on paraffin should not exceed 65 to 70 lb.* per square inch and that magneto ignition is preferable to coil and battery ignition. They admit also that a drop in horse-power in the engine must be expected of 10 to 15 per cent. compared with the horse-power obtainable when using petrol. This loss is chiefly due to the higher temperature of the entering fuel which with a given cylinder volume naturally reduces the weight of charge admitted. There is also a loss owing to the necessary lowering of the compression ratio and consequent lowering of thermal efficiency. This lowering of the compression pressure is due to fear of back-firing (through the inlet ports before they have quite closed) caused by the charge being pre-ignited owing to the temperature rising to the point at which paraffin vapour and air ignite spontaneously. Lowering the compression pressure naturally lowers the compression temperature.

There is a gain, however, in that the calorific value of paraffin per pound is about 18,000,000 ft.-lb. against about 15,000,000 for petrol; also an advantage is found in the smaller consumption of lubricating oil owing to the lubricating properties of the paraffin itself.

As petrol costs two or three times as much to buy as paraffin it is clear that even when allowance is made for a drop in horse-power there is still a considerable gain in economy from the financial point of view. The patentees

*Many engines work at 80 lb. per sq. inch compression.

of the Cottrell carburettor have published the following comparison of costs of the two methods of working—

| Pleasure Vehicles. Horse-power. | Price Petrol and Paraffin (per gallon). | Cost per Mile. | Miles average per gal. | Assumed Mileage per Annum. | Mileage to be run to Cover Cost of Carburettor. |
|---|---|---|---|---|---|
| 6 h.p. single cylinder . | 1/2 & 6*d.* | ·4 / ·17 | 35 | 8,000 | 4,460 |
| 10 ,, double ,, . | 1/2 & 6*d.* | ·46 / ·20 | 30 | 8,000 | 6,500 |
| 15 ,, four ,, . | 1/2 & 6*d.* | ·56 / ·24 | 25 | 9,000 | 7,000 |
| 30 ,, ,, ,, . | 1/2 & 6*d.* | ·7 / ·3 | 20 | 10,000 | 6,500 |
| 50 ,, ,, ,, . | 1/2 & 6*d.* | ·93 / ·4 | 15 | 10,000 | 4,800 |
| 100 ,, ,, ,, . | 1/2 & 6*d.* | 1·75 / ·75 | 8 | 10,000 | 3,300 |
| Commercial Vehicles. | | | | | |
| 10 h.p. single cylinder . | 1/– & 4½*d.* | ·48 / ·18 | 25 | 50,000 | 3,500 |
| 20 ,, double ,, . | 1/– & 4½*d.* | ·6 / ·225 | 20 | 50,000 | 4,200 |
| 20 ,, four ,, . | 1/– & 4½*d.* | ·8 / ·3 | 15 | 45,000 | 4,550 |
| 50 ,, ,, ,, . | 1/– & 4½*d.* | 1·2 / ·45 | 10 | 45,000 | 3,600 |

Figures such as the above depend very largely upon the fuel consumption per mile run by the car. From tests carried out by the author it appears that a 20 h.p. transport car weighing about 2 tons and carrying 2 tons, will run eleven miles on one gallon of petrol, and ten miles on one gallon of paraffin when using the Cottrell carburettor as illustrated.

**118. Milnes-Daimler Carburettor.**—The Milnes-Daimler firm have recently introduced a new form of carburettor which is claimed to be equally suitable for petrol, alcohol or paraffin. It has a jet and float-feed of the usual type. The air passes the jet in an upward vertical direction, not a horizontal one. The working of the instrument is best described by stating what happens to the air from the moment of admission until it is delivered to the cylinder. Air

is drawn first of all into a jacket surrounding the exhaust; this raises the temperature to a considerable degree, and on leaving the jacket the air pipe has holes in it controlled by a rotating sleeve of the usual form in order by admitting more air to reduce the temperature to the requisite extent. If petrol is to be used a good deal of cold air is admitted; if paraffin very little, if any, additional air is required. The air then passes upwards around the jet. The jet itself has a hollow truncated cone above it (the angle of the sides being about 15°) so that if the cone is lowered the air passage is reduced in area. In this way the top of the jet is in the middle of an annular space through which air flows and the

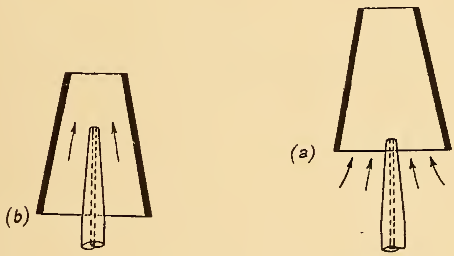



FIG. 93.—Positions (*a*) and (*b*) of Hood in Milnes-Daimler Carburettor.

area of this annular space can be varied in a ratio of about 1 to 2. The idea is to increase or decrease this area so that the velocity of the air past the jet may be kept from varying.

The "truncated cone" sleeve is raised and lowered by the governor and it is made in one piece at the top with a thin brass sliding cylinder having ports in it. These ports are opened as the sleeve rises and extra air (cold) is taken in. The whole gaseous mixture then passes out through upper ports in the same cylindrical shell, which being controlled by the governor causes the necessary throttling. The governor, therefore, not only works the throttle but also the extra air inlets and the cone sleeve just above the jet. Adjustment for different densities and qualities of petrol can be made by slightly rotating the cylindrical shell and then fixing the position by screws. An engine

fitted with this type of carburettor is tested on the bench and set so as to give greatest petrol, or other fuel, economy at the normal speed, say 800 revolutions per minute. If the governor really could by its adjustments of the conical sleeve keep the air velocity constant the rate of flow of petrol would be constant, and the richness of the air mixture immediately after passing the jet would be inversely proportional to the effective area of the air passage in the annular space around the jet. The extra air is admitted to keep the quality of the mixture the same. All this depends on the action of the governor, that is to say on the engine speed, and its successful working depends on a proper shape for, and adjustment of, the various openings and ports.

**119. Another type of paraffin carburettor** is the "Broom & Wade." Here the paraffin is supplied to a valve type of inlet—as shown in Fig. 82, and enough air is admitted to get the requisite quantity of fuel to enter and scrub its way past several layers of wire mesh which serve to break up the liquid. On the suction stroke the valve opens to admit this to the cylinder, and later on in the stroke a valve opens which admits through an inlet in the cylinder wall the requisite extra air to make the mixture correct. The firing point (low tension magneto) is well back from the cylinder, at the end of a sort of combustion chamber which is made in one with the carburettor (an arrangement which makes a separate jacketing of the carburettor by exhaust gases unnecessary). The engine is started up on petrol, then changed over to paraffin, and the control is through the medium of a lever which affects the lift of the suction valve. At heavy loads there is likelihood of pre-ignitions owing to the high temperatures reached, and it is therefore arranged that water can be admitted with the extra air, the water being carried in past the valve in much the same way as the paraffin is carried past the suction valve.

**120. The Thornycroft type of paraffin carburettor** is illustrated in Fig. 94. Its manner of working is generally similar to that of the Cottrell, but the heating surface is

less in proportion. It works well in practice and its mode of operation is as follows :—

The oil is drawn into the vaporizer together with a certain amount of air by the suction of the engine; this mixture is then passed through a tube which is kept heated to a fairly high temperature by the exhaust gases coming from the engine. This thoroughly vaporizes and intimately mixes the vaporized oil and air. The mixture is then passed through a spiral separator, which separates any solid matter from the vapour, is mixed with extra air as required to form an explosive mixture, and then passed through the throttle to the cylinders. In the drawing, *A* is the

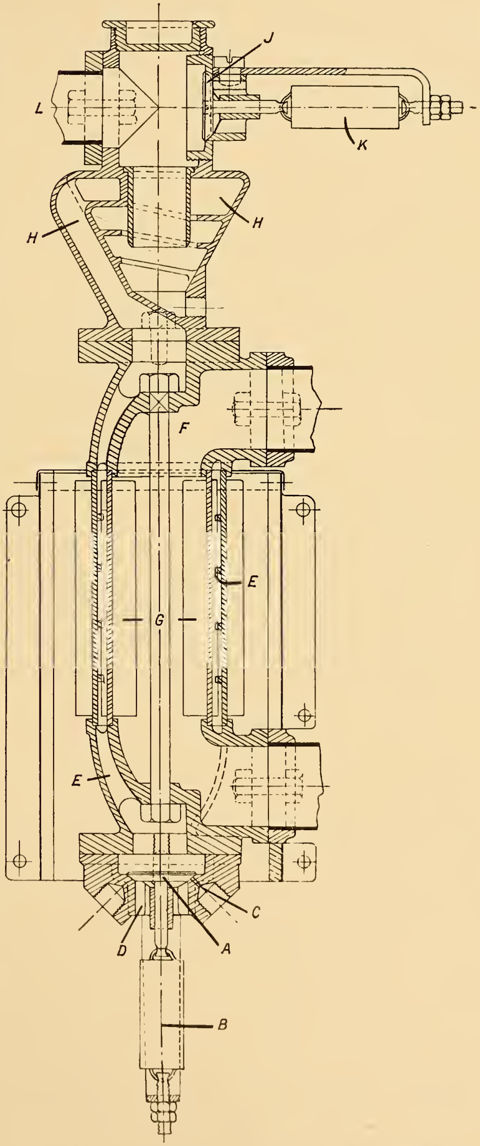



Fig. 94.—Thornycroft Paraffin Carburettor.

inlet valve for both oil and air, the valve being under

the action of spring $B$, which normally keeps the valve closed and the oil supply $C$ shut off; the oil enters by holes in the seating. The mixture then passes along the annular space $EE$ which is kept heated to a high temperature by the exhaust flowing through the centre of this annular chamber as shown at $F$. The annular chamber it will be noticed is fitted with gills $G$ to enable a maximum amount of heat to be supplied to the mixture. $H$ illustrates the separator for removing the solid particles from the mixture, and $J$ the "extra-air" inlet, under the control of the spring $K$. The tension of this spring and also of that governing the inlet of the mixture to the vaporizer can be varied by a screw and nut as shown. This adjustment is made when the engine is on the test-bed before the brake trials. The outlet from the vaporizer to the motor is by pipe $L$.

**121. Theory of Jet Carburretors.**—The energy equation for the flow of any fluid (liquid or gas) is as follows—*see* Perry's *Applied Mechanics*, p. 533—

$$\frac{v^2}{2g} + \int \frac{dp}{w} + h = \text{constant} \quad \ldots \ldots \ldots \quad (1)$$

This is true for any stream line.

$v$ = velocity of fluid.
$g$ = 32·2.
$p$ = pressure.
$w$ = weight of unit volume or density.
$h$ = potential energy due to height.

If this equation be applied to the flow of air through the carburettor due to the suck of the engine, it may be simplified in many ways. We want to find the amount by which the pressure in the rushing air is made lower than the atmospheric pressure owing to the suck of the engine pistons. When the air is at rest equation (1) becomes

$$\int \frac{dp}{w} + h = \text{constant} \quad \ldots \ldots \ldots \ldots \quad (2)$$

when air is rushing with velocity $v_0$ then

$$\frac{{v_0}^2}{2g} + \int \frac{dp}{w_0} + h_0 = \text{constant} \quad \ldots \ldots \ldots \quad (3)$$

Now $h$ and $h_0$ are the same, as the air may be taken to flow horizontally.

Therefore equating (2) and (3) we have

$$\int \frac{dp}{w} = \frac{{v_0}^2}{2g} + \int \frac{dp}{w_0} \quad \ldots \ldots \ldots \ldots \quad (4)$$

but if unit weight of air be considered, $w = \frac{1}{V}$, where $V$ is volume,

and $pV^{\gamma}=$ constant is the approximate law connecting $p$ and $V$ when no heat is given or received. $\gamma = 1{\cdot}41$ as usual. Therefore the expression $\int \frac{dp}{w}$ becomes

$$\int V.dp$$

$$= \int \left(\frac{c}{p}\right)^{\frac{1}{\gamma}} dp$$

$$= c^{\frac{1}{\gamma}} \int \frac{dp}{p^{\frac{1}{\gamma}}}$$

$$= \gamma c^{\frac{1}{\gamma}} \frac{p^{1-\frac{1}{\gamma}}}{\gamma - 1}$$

Substitute in equation (4) and then we have

$$\frac{v_0^2}{2g} = \frac{\gamma}{\gamma - 1} c^{\frac{1}{\gamma}} \left\{ p^{\frac{\gamma-1}{\gamma}} - p_0^{\frac{\gamma-1}{\gamma}} \right\}$$

Let difference in $p$ and $p_0$ be called $\delta p$, then since $\delta p$ is small

$$\frac{v_0^2}{2g} = \frac{\gamma}{\gamma - 1} c^{\frac{1}{\gamma}} \left\{ (p_0 + \delta p)^{\frac{\gamma-1}{\gamma}} - p_0^{\frac{\gamma-1}{\gamma}} \right\}$$

$$= \frac{\gamma}{\gamma - 1} c^{\frac{1}{\gamma}} p_0^{\frac{\gamma-1}{\gamma}} \left\{ \left(1 + \frac{\delta p}{p_0}\right)^{\frac{\gamma-1}{\gamma}} - 1 \right\} \text{ approximately.}$$

which reduces to

$$\frac{v_0^2}{2g} = \frac{\delta p}{w_0} \quad \ldots \quad (5)$$

From this we have

$$\delta p = \frac{w_0 . v_0^2}{2g} \quad \ldots \quad (6)$$

The pressure at the top of the petrol jet is therefore lower than the pressure on the surface of petrol in the float chamber by $\delta p$ where $\delta p$ is given by equation (6). A somewhat similar line of argument to the above will show that the flow of the petrol is determined by a formula of the same type. For a liquid such as petrol in which $w$ is independent of $p$, equation (3) becomes

$$\frac{v^2}{2g} + \frac{p}{w} + h = \text{constant} \quad \ldots \quad (7)$$

Applying this equation to the stream line which begins at the free petrol surface in the float chamber and ends in the jet spray and assuming that the top of the metal jet is a height $h$ above the petrol level (so that petrol has to rise through a height $h$ in the jet before it gets to the actual nozzle) we have

$$o + \frac{\delta p}{w} + o = \frac{v^2}{2g} + o + h$$

where $v =$ velocity of flow of petrol and $w =$ density of petrol (i.e. the weight per cubic ft.).

Therefore $$\frac{v^2}{2g} = \frac{\delta p}{w} - h$$

Now by equation (6):— $$\delta p = \frac{w_0 v_0{}^2}{2g}$$

so that $$\frac{v^2}{2g} = \frac{w_0 v_0{}^2}{w2g} - h$$

or $$v^2 = \frac{w_0}{w} v_0{}^2 - 2gh \quad \ldots \ldots \ldots \ldots \quad (8$$

This shows that $v_0$, the velocity of air flow, must have the value $\sqrt{2gh \frac{w}{w_0}}$ before any petrol will flow at all. It is a matter of common experience that if the rate of air flow doubles the petrol flow more than doubles. Let us see if this is so according to this formula.

First, let $v_0 = a$, and then equal $2a$. The ratio of the square of the petrol flow in the second case to that before should therefore, according to experience, be more than 4—by formula (8) it is

$$= \frac{\frac{w_0}{w} 4a^2 - 2gh}{\frac{w_0}{w} a^2 - 2gh}$$

$$= 4 + \frac{6gh}{\frac{w_0}{w} a^2 - 2gh} \quad \ldots \ldots \ldots \ldots \ldots \ldots \quad (9)$$

which it will be seen does exceed 4, as experience records.

It will be interesting to get some quantitative figures for this. What, for instance, will be the ratio if $h = 0{\cdot}04$ feet (or $\frac{1}{2}$ inch) and $a$ is 5,000 ft. per minute ?

Petrol has a density of about 0·72 so that 1 cu. ft. will weigh $0{\cdot}72 \times 62{\cdot}3 = 45$ lb. Whereas 1 cu. ft. of air weighs 0·085 lb. at atmospheric temperature and pressure.

According, therefore, to equation (9) the square of the ratio of the two petrol velocities will be

$$4 + \frac{6 \times 32{\cdot}2 \times 0{\cdot}04}{\frac{0{\cdot}085}{45} a^2 - (64{\cdot}4 \times 0{\cdot}04)}$$

$$= 4 + \frac{7{\cdot}72}{\frac{a^2}{530} - 2{\cdot}57}$$

The critical velocity is clearly

$$v_0 = \sqrt{2gh \frac{w}{w_0}} = \sqrt{2 \times 32{\cdot}2 \times 0{\cdot}04 \times 530} = \sqrt{1370} = 37{\cdot}0 \text{ ft./sec.}$$

$= 2{,}220$ ft. per min. Until, therefore, the air had this velocity no petrol would be carried along. Since $a = \frac{5000}{60}$ the equation (9) becomes

$$4+\frac{7\cdot72}{13\cdot0-2\cdot57}=4+\frac{7\cdot72}{10\cdot4}$$
$$=4\cdot74.$$

So that when air velocity increases from 5,000 to 10,000 ft./min. the petrol velocity is 2·18 times as much and the mixture therefore 1·09 times as rich or 9 per cent. richer. This means that the amount of petrol present per cubic foot of air increases by one-eleventh part. This inequality is really most marked when the velocity of the air is only a little more than what is necessary to feed the petrol, thus if the air velocity be increased from 2,500 ft./min. to 5,000 ft./min. the petrol sucked along would be increased by no less than 290 per cent., i.e. the quantity of petrol would be 3·9 times as great, giving a richness of mixture 1·95 times or nearly double what it was before. In this case, therefore, about twice the quantity of air would be needed, i.e. as much again must pass the additional air inlet as already passes the jet. This theory accounts for part of the extra air needed. It does not, however, take account of the effect of eddies, that may circulate around the jet at high air velocities, and so complicate the problem a great deal.

**122.** It is possible that there are still **further reasons** why extra air is needed when the engine speed increases, but the above are certainly some of them. The air velocity is highest when the engine is running fast and the throttle wide open. When the engine is running fast but is much throttled, as in a car running very fast down a slope, the vacuum behind the piston never gets filled up with air and the velocity of air past the jet is not therefore very great. Going up hill, however, on low gear the engine speed is high and the throttle wide open; so that the velocity of air past the jet is not solely dependent on the engine speed. This makes control of the additional air inlet by the centrifugally operated governor not as uniformly good as could be wished—it only approximates to extreme cases, fitting accurately the average only. When, however, as in the Krebs carburettor, for instance, the opening of the extra air inlet is controlled by the suction a much more constant mixture is obtained. A carburettor is usually set so that the right mixture comes away from the jet at low speeds with the extra air inlet closed. Then as the speed rises additional air is allowed to pass in. From the preceding calculations it will be clear that one cause of the lack of proportionality between the air and petrol velocities is the fact that the petrol cannot be allowed to stand at a level equal to the height

of the jet, as if this condition were arrived at too closely there would be a risk of the petrol overflowing and running away to waste. A further reason which makes it necessary not to run the adjustment too finely is that the level of petrol in the float chamber is bound to vary somewhat not only with the inclination of the float chamber but also with temperature and quality of supply. In order to keep on the safe side the petrol level must be a good many millimetres below the top of the jet.

**123. Ignition.**—Types of ignition are many, and the author proposes to describe several examples of modern methods, but before doing so it is necessary to say something about the principles upon which they work. The oldest form of all was to ignite the explosive mixture by a naked flame, which was put into communication with the cylinder through the medium of a sort of slide valve. It is quite obsolete now, but those interested in the history of the subject will find full details in Mr. Dugald Clerk's book.

A later and much more successful form was **Tube ignition,** which consisted in having a short vertical tube, in communication with the cylinder end, heated externally by means of a lamp. After it had been running a short while the lamp could be removed (or the gas jet turned out) and the heat of ignition was enough to keep the temperature up to the requisite point. It is illustrated in Fig. 95. The explosive charge is compressed by the inward movement of the piston, and a part of it passes into the ignition tube, the temperature of which raises the temperature of the gases at the end of the stroke to just the correct temperature for explosion. It may seem as though it would be difficult to effect such a nice adjustment as this, but it works more successfully than might be expected. It appears as though the criterion that settled the moment of explosion, provided the circumstances were favourable, was the movement of the gases to expand just at the instant when the crank passes the dead centre and the piston begins its outward journey. This type of ignition has been very largely used for gas engines, but it is much less common now. It was also the first method employed on motor cars,

for which use it was obviously unsuitable and speedily fell into disuse.

Another method of ignition often used with oil engines is to feed the fuel right into a hot combustion chamber connected to the cylinder. This method as carried out on the Hornsby engine has already been illustrated. It works very well and even crude Borneo oil can be vaporized and ignited in this way. The method employed on the Diesel engine is of the same general type.

**124. The chief method of ignition is the electric,** and it bids fair to supersede all the others—indeed it has

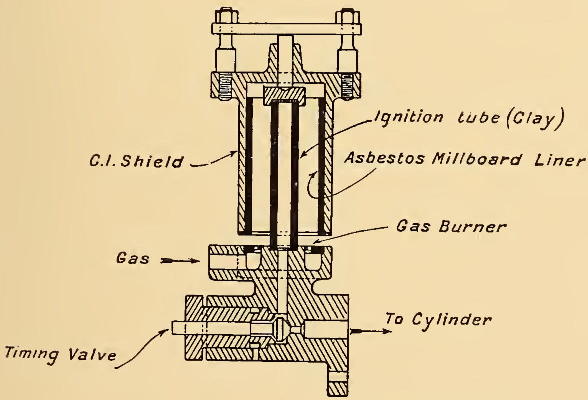



FIG. 95.—Ignition Tube and Timing Valve.

nearly done so already. It is equally suitable for gas, oil or petrol engines.

Electric ignition can be carried out by either (1) **high-tension** currents or (2) **low-tension** currents. In the former the voltage runs into many thousands of volts, and in the latter probably only into hundreds. The word "probably" is used, as unless an oscillograph is employed, it is very difficult to say exactly what the voltage is.

The high-tension currents may be obtained in three ways: either (1) by batteries or cells furnishing current to an induction coil, or (2) by a small magneto-electric machine (called "magneto" for short) furnishing currents to an induction coil, or (3) by a magneto furnishing high-tension

currents direct to the sparking plug (the name given to the spark gap in the cylinder). The low-tension currents are produced from a low-tension magneto. There are therefore many varieties of electric ignition and they may conveniently be set out thus—

- Electric Ignition
  - High tension
    - Batteries or cells with induction coil
      - Coils fitted with tremblers (1)
      - Coils fitted with fixed "make-and-break" (2)
    - Magneto with induction coil (3)
    - High-tension magneto (4)
  - Low tension
    - Low-tension magneto (5)

This makes quite a family tree and it is clearly a case in which an engine builder has a considerable variety of choice. There are five main varieties. The oldest is (1) and it is still seen fitted to some petrol engines, although (2) is more common; (3) is relatively rare but is seen in the Eisemann system; (5) is common practice and a most satisfactory method which is coming more and more into use. It is particularly suitable for engines which have to work in a tropical or sub-tropical climate. Method (4), however, is growing in popularity and it has the advantage of being simpler to apply to the engine than (5).

Before describing methods (1), (2) or (3) it will be necessary to say something about the induction coil. To those who are versed in electrical matters it is enough to describe it as a transformer having a straight iron core and a high ratio of transformation. To others not so familiar with electrical matters a little more explanation will be necessary.

**125. Induction Coil.**—The induction coil used on a car is similar to the usual type. It consists of a soft iron core generally consisting of a bundle of straight iron wires and on it is wrapped a layer or two of thick insulated copper conducting wire of the primary—or low-tension—circuit. Over this

is wound many thousand turns of fine insulated copper wire constituting the secondary—or high-tension—circuit. About 4 volts are applied to the primary circuit and the current repeatedly broken and remade by means of the magnetism of the iron core attracting a small piece of iron mounted on a spring which carries the current. As the spring is attracted inwards it loses contact with a platinum point and so breaks the current. (To make the break the more sudden it is usual to put a condenser in parallel in the circuit.) This sudden rise and fall of current in the primary causes oscillatory currents in the secondary of a voltage which is higher than that in the primary in the ratio of the number of coils of wire in one to the number in the other. Owing to the effect of the magnetism in the iron core the current in the primary does not rise suddenly to its full value. It follows in fact the law

$$C=\frac{V}{R}\cdot\left(1-\epsilon^{-\frac{R}{L}t}\right)$$

where

$C$=current in amperes.
$V$=voltage in volts.
$R$= resistance in ohms.
$L$=self-induction in henries.
$t$=time in seconds.
$\epsilon$=base of Naperian logarithms or 2·7183.

The unit of self-induction is the henry. If $S$ be the rate at which the current changes in amperes per second, the back electro motive force produced $=L\times S$. One henry is also defined to be the self-induction of a coil in which, if the current increase at the rate of one ampere per second, the back E.M.F. produced is exactly one volt.

As an illustration of the effect of the above law of rise of the current, take the case of a coil in which $R=500$; $L=5{\cdot}5$ and $V=50{,}000$. Then the final and steady value of the current is clearly $\frac{50{,}000}{500}$ or 100 amperes. This current grows up from zero and it is of interest to calculate how long it will take 90 amperes to be the current flowing.

$$90=100\left(1-E^{-\frac{t}{0\cdot011}}\right)$$

or $$E^{-\frac{t}{0\cdot011}}=1-0\cdot9=0\cdot10$$

so that $t$=about $\frac{1}{35}$ second.

The current therefore rises by no means *instantaneously* and this leads to the "make" of the primary current, producing a much less vigorous spark in the secondary than does the "break." The break is almost instantaneous; the only thing that tends to prevent it being so, is the energy of rush of the primary current which jumps over the gap in its earlier stages. The energy stored up in the flowing current is equal to $\frac{1}{2}LC^2$, and it is to provide a convenient swamp to absorb this suddenly released energy that the condenser is provided.

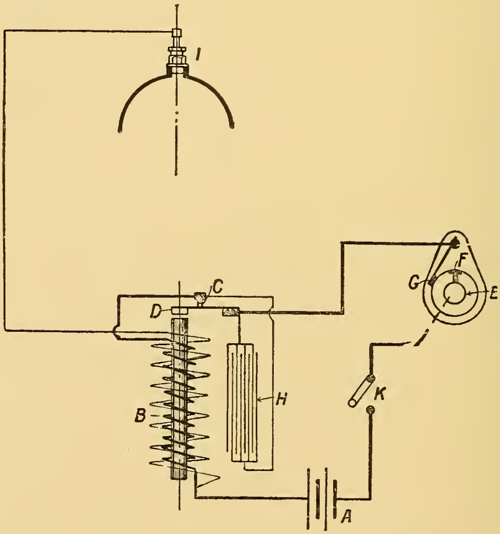



FIG. 96.—Diagram showing mode of working of high-tension ignition with coil and accumulator. *A*, Accumulator. *B*, Induction coil. *C*, Contact breaker. *D*, Trembler. *E*, Commutator on end of cam shaft, for closing circuit at right moment by bringing metal segment *F* against the brush *G*. *H*, Condenser, to make break of current sudden. *I*, Ignition plug in cylinder. The other end of the secondary winding is earthed.

**126. High-Tension Coil Ignition.**—The induction coil is supplied with a low-tension current obtained from either

batteries, accumulator cells or a magneto. In any case the principle of working is the same. The spark gap is placed in the cylinder as shown in Figs. 96, 97, 98, and 99.

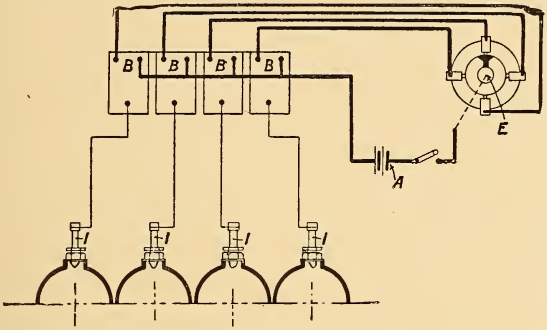



FIG. 97.—Coil and accumulator Ignition, for a four-cylinder engine, with separate coils for each cylinder. *A*, Accumulator. *B*, Coils each with its own condenser and contact maker. *E*, Commutator for distributing current to the various cylinders at the right moment. *I*, sparking plugs.

A rotating contact kept at a speed proportional to that of the engine and called a distributor distributes the current to each cylinder just as it is needed. What happens there-

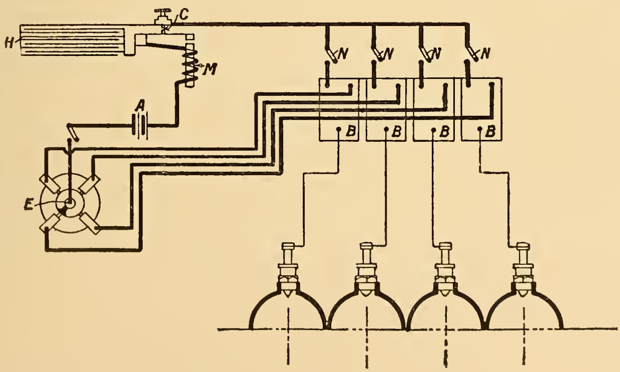



FIG. 98.—The arrangement shown in Fig. 97, except that one trembler serves all the coils. This saves having to adjust each trembler until all are working at same frequency. *C* is the common contact maker, and *N* are switches for cutting out coils when necessary.

fore is this. The trembling blade on the coil—called the trembler—vibrates very rapidly and produces a shower of sparks in the secondary (one spark corresponding to each

break of current in the primary). During each contact about a dozen sparks or more may pass. One good spark would be enough and therefore a modification of this method is sometimes employed. Instead of a trembler actuated by the magnetism of the iron core of the coil, the current in the primary circuit is made and broken by the action of the engine. A mechanical make-and-break is

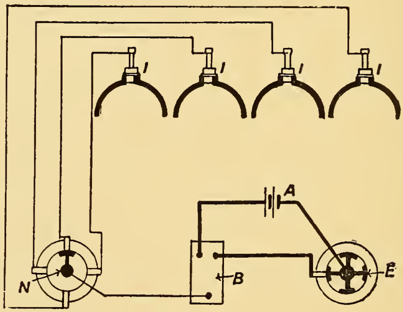



Fig. 99.—The arrangement of Figs. 97 and 98, except that the secondary current is distributed directly, so enabling only one coil to be used for all four cylinders. The disadvantage is that the insulation is more difficult to ensure.

fitted to the half-speed shaft of the engine so as to produce one spark only in the cylinder. It is possible to ring the changes on this form of ignition so as to produce a great many varieties, although the differences between them are hardly fundamental. Illustrations, reproduced from Mr. Strickland's useful book, are shown of several such methods (*see* 96, 97, 98, and 99), and the letterpress at the foot of each will suffice to show their differences.

**127.** The **Lodge** (Sir Oliver Lodge) system of ignition is just the ordinary coil and accumulator ignition in which the high-tension current instead of being passed direct to the sparking plugs is made to charge up the innner coatings of two Leyden jars. When the jars are " full " an external spark gap placed in parallel with the jars breaks down and a spark passes. This sudden release of the electric charges on the inner coats of the Leyden jars causes such a rush of current from the outer coating of one Leyden jar to the other and such a violent oscillation to and fro of the current

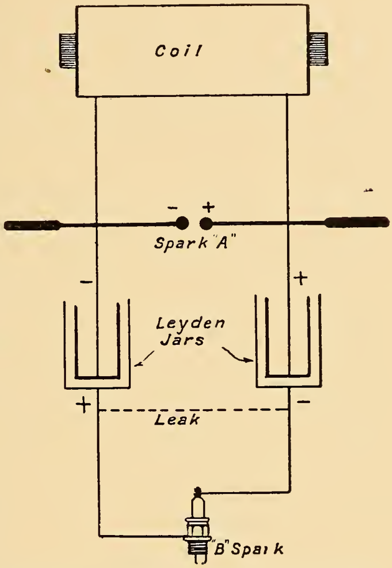



FIG. 100.—The Lodge Ignition System. Diagram of H.T. circuits.

afterwards that nothing will stand in its path. It breaks through oil films, soot, deposit of all kind, water or anything else that there may be on the ignition points; owing to its high frequency it also tends to take straight direct courses, and there is little disposition on its part to seek any short circuit of a tortuous kind which may happen to be in existence.

Fig. 100 shows diagrammatically the arrangements of the high-tension circuit. The low-tension circuit is of the customary form, except that the trembler shown in Fig. 101 is of an extra sensitive form. The distributor is placed in the high-tension circuit. The makers of this ignition system claim that owing to the adjustments made no possible error in the time of firing can arise which exceeds $\frac{1}{3000}$th part of a second. Also that in virtue of the nature of the spark the system is particularly suitable for use when the fuel used is paraffin or any other heavy oil which may cause carbonaceous deposit on the ignition plugs.

**128.** We now come to **magneto ignition.** It may be either *high tension* or *low tension.* No coil is used and no batteries or cells are wanted. The low-tension method proceeds on the principle that when a current is flowing it has

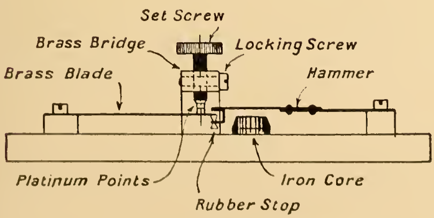



FIG. 101.—Lodge Sensitive Trembler.

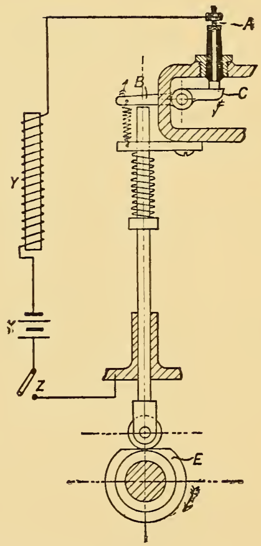



FIG. 102.—Low-tension magneto ignition. *X* and *Y*, magneto machine shown diagrammatically. *A*, Spark plug. *C*, Contact point where circuit is closed and broken. *B*, Lever worked by rod running on cam, *E*. *E*, Cam on half-time shaft. At the moment when the magneto is passing its maximum current around the circuit, the cam causes the circuit to be broken at *C*, so producing a spark at that point.

energy of motion equal to $\frac{1}{2}LC^2$ (analogous to kinetic energy, $\frac{1}{2}mv^2$), and that if $L$, the self-induction, is made very great and $C$, the current, as great as convenient, the energy stored up is so considerable that a large or "fat" spark is caused to occur when the circuit is suddenly broken. A low-tension magneto, or electric generator, is designed so as to cause such a current to be passing at the moment when ignition is desired to occur and, at the same instant, the circuit is mechanically broken in the cylinder and a spark passes. Fig. 102 shows diagrammatically the wiring for this system and the sparking plug used. A larger view of such a plug is seen in Fig. 103.

The high-tension magneto consists of a magneto in which a number of turns of fine wire are wound on to the armature so as to act as a secondary circuit. The effect is that very high voltage currents are produced and the current can be led to sparking plugs of the ordinary type in the cylinders. This has

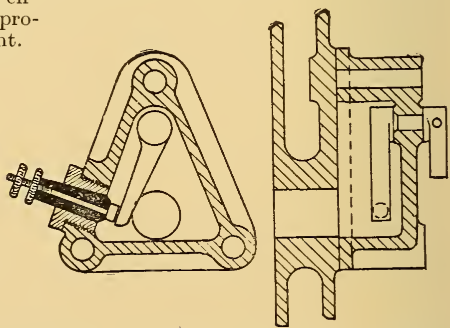


FIG. 103. — Typical low tension magneto spark plug. It will be noticed that this system of ignition requires moving contacts in the cylinder, which the high tension system does not.

the advantage that no moving parts need to be introduced into the cylinder in order to produce a spark. The voltage is so high that a "safety valve" spark-gap is usually fitted in parallel near the magneto in order to allow any unduly high voltage current which may be produced to pass across it. The spark in this gap is, of course, of no use except to act as a safety valve or bye-pass.

**129. The Simms-Bosch High-Tension Magneto System of Ignition.**—In this system the current is generated by a shuttle armature which rotates between the poles of three pairs of very strong steel magnets. The rotation of this armature in the strong magnetic field results in the induction in its winding of an electrical current which is utilized for the purpose of ignition. The armature is wound in two parts,

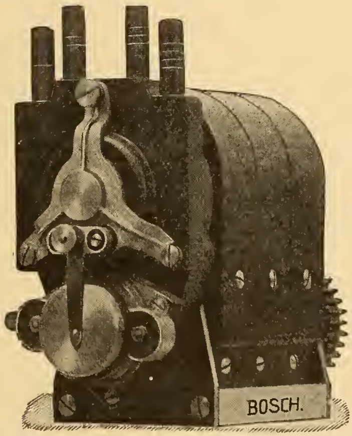



FIG. 104.—Outside view of Bosch H.T. Magneto. (A low-tension magneto is of generally similar shape.)

of which one is a primary winding, consisting of few turns of heavy wire, and the other a secondary winding, consisting of many turns of fine wire. The effect is that a high-tension current is given off by the armature, as the design practically amounts to the inclusion in the armature of the windings of an induction coil. An outside view of this magneto is shown in Fig. 104, and in the appendix to this chapter a full description is given of its manner of working.

**130. Timing of Ignition.**—One of the most careful adjustments of the ignition is its timing. That is to say, the regulation of the moment of sparking in the cylinder. If the spark is late the piston has moved part of its outward journey, with the consequence that the effective working stroke is lessened and the mean pressure is much lower than it should be. If the spark is too early, so that the gases are still being compressed when the spark comes, then there is a knock in the bearing when the explosion occurs. Normally the spark should occur just as the piston is at the top of its stroke, although since ignition takes a fraction of a second to spread throughout the mass of the gas it is necessary when the engine is running fast to time the spark to occur a little before the dead centre so that maximum pressure is reached when the piston is just beginning its stroke. Engine speed is, however, not the only consideration affecting the timing; when running with weak mixtures the ignition takes longer than with rich mixtures so that to use a weak mixture it is necessary to "advance" the spark, i.e. make it occur earlier. It follows, therefore, that in the ordinary running of a car the ignition requires as much attention as the throttle, if the engine is to work at highest efficiency. An additional complication arises when coils having tremblers are used with batteries or cells, as the speed of "trembling" being naturally independent of the speed of the engine, it follows that unless the ignition is advanced with increase of engine speed the sparking in the cylinder will occur later in the stroke. Cars—chiefly heavy ones—fitted with low-tension magneto ignition sometimes run on fixed ignition, i.e. the spark is designed to occur as near as possible always to the dead centre. This, however, may make it difficult to start the engine, since if the spark occurs before the instant of dead centre the pressure will produce a heavy blow on the starting handle and possibly break the driver's wrist. It is claimed sometimes that with low-tension magneto ignition the ignition is advanced or retarded with the speed of the engine almost automatically, owing to a "fatter" spark being produced

when the magneto is running fast and a presumed increase in speed of ignition. In some engines it is arranged that the governor shall control the ignition, and this plan is adopted in the Albion engine.

**131.** The Albion engine is fitted with a low-tension magneto system of ignition and Fig. 105, which is reproduced from a paper * by Mr. T. Blackwood Murray, shows very clearly the **effect of mechanical lag** in the movements of the mechanism at high speeds. As the speed increases so the firing point is advanced, until at 1,000 revolutions per minute

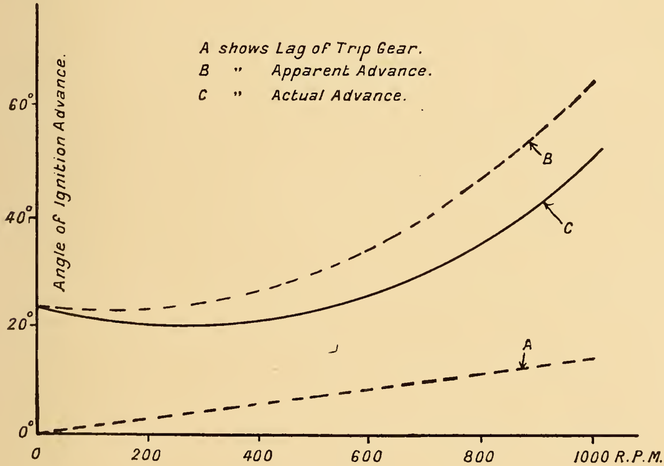



FIG 105.

it is adjusted so that the spark would occur no less than 64° before the dead centre, but owing to a 14° lag on the part of the trip gear the spark only comes 50° early. In the Albion engine this control is left to the governor entirely. It will be noted that even at starting there is a real advance of about 20°, and this might be expected to make the starting rather difficult—indeed dangerous—to an unskilled driver, but from an experience of many Albion engines the author has never noticed any difficulty to arise from this cause. Mr. Murray's view of the matter is that :

* "Some Details of Albion Motor Cars," read before Institution of Engineers and Shipbuilders in Scotland, 1907.

"Owing to the fact that no spark can take place unless the crankshaft is rotating with a certain angular velocity, it is permissible to set the ignition to take place, even at starting, slightly before the dead centre, as this said velocity ensures the *vis viva* of the flywheel carrying the engine over the dead centre. This reduces the arc of ignition advance throughout which the magneto is called upon to generate an effective spark, and enables one to key the magneto once for all in a fixed position to the crankshaft, which might not be possible if too large an arc of advance were necessary." In other words, it has been sought to find a compromise between conflicting ideals of operation.

In the Albion method, between fast and slow speed the point of ignition is moved through about 30° of the circumference of the crank path. Thirty degrees at 1,000 revolutions per minute corresponds to a time interval of $\frac{60}{12,000} = \frac{1}{200}$ second, and this extra interval it is which allows the ignited gases to be at their full pressure just as the highly speeded engine arrives at the dead centre. The ignition obviously affords a method of governing the engine, as if the spark be made to occur very late in the working stroke hardly any horse-power will be produced, but this plan is expensive in fuel and it is better to decrease the horse-power by decreasing the volume of mixture admitted to the cylinder. Here we have written rather of the ignition as affecting petrol engines than as affecting gas engines, large or small. The reason is that a petrol engine, especially as used on a car, is far more sensitive to such changes and makes a much better exploring instrument than a gas engine. But what applies to the one applies also to the other. Magneto ignition is becoming more and more common on gas engines, especially low-tension ignition, and what is still more striking is to see that gas engine builders are taking a leaf out of the books of the petrol engine builder and making the ignition variable so that it may be adjusted to starting conditions and sometimes even to exceptional running conditions.

**132.** In a paper read by Dr. Watson before the Royal

Automobile Club an interesting account was given of certain experiments undertaken to ascertain the **character of the spark in relation to power.** The engine used was a two-cylinder one, 3·5 in. ×4 in., with mechanically operated valves. The sparking plug was screwed into the cap used to close the hole over the inlet valve, the spark points being well inside a recess in this cap. The whole of the experiments were made on one cylinder only, the other being operated with a trembler coil and battery. The speed was 950/1,000 revolutions per minute. It has often been claimed that a "fat" spark improves the running, and that this was due either to quicker ignition of the charger or to more regular firing. Experiments with a trembler coil showed that although the weakening of the current was found to reduce the mean pressure, yet this *could be brought back to its original* value by advancing the spark. The result of this series of experiments was to lead Dr. Watson to the following conclusions—

1. As far as a petrol engine of the type used is concerned, the character of the spark which ignites the charge has no appreciable influence on the power developed.

2. With a trembler coil the time at which the spark occurs is liable to vary greatly, and on this account the power developed may be considerably reduced.

3. The variation in the time of firing obtained with trembler coils is different for different coils, and hence a multi-cylinder engine in which a separate coil is used for each cylinder is unlikely to develop its maximum power, particularly at high speeds ; the reason being that although the tremblers of the coils may possibly be so adjusted for some particular voltage that each cylinder fires at the same point of the stroke, yet this adjustment will no longer be true if the voltage of the battery alters, particularly if it falls much below the value for which the tremblers were adjusted.

4. When a single coil is used in combination with a high-tension distributor, it is of very great importance that the current in the primary should never be allowed to fall to a value near the critical value for the particular coil. In this connexion it may be mentioned that, in Dr. Watson's experience, when the trembler is so adjusted for any given voltage of the battery, i.e. for a given current, that the note produced is very clear and "pure," then a very slight decrease in current, due to a small fall in the voltage of the battery, will cause the timing to be defective, owing to the region of the critical current being approached. Hence, with the normal current passing—i.e. with the battery fully charged—it is advisable to adjust the trembler so as to give a somewhat harsh and shrill sound, for

then the current may be considerably reduced before the critical value is reached.

5. When selecting a coil, regularity in the working of the trembler for considerable variation in the current passing in the primary is of more importance than length or fatness of spark. Further, a coil taking a small current is to be preferred to one taking a large current, since trouble with the adjustment of the trembler blade will be decreased, owing to the reduced sparking at the platinum points with a small current.

6. Except for the fact that the engine cannot be started on the switch, the plain coil with a rapid break on the two-to-one shaft

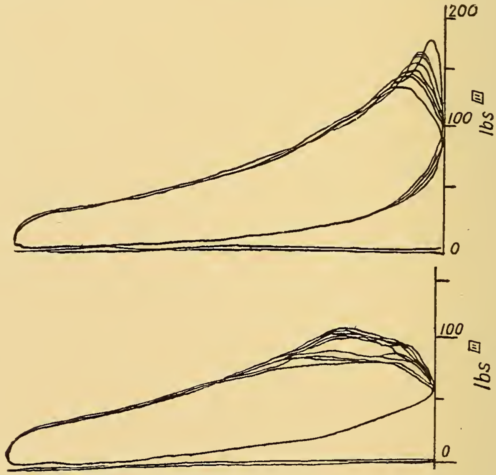



FIG. 106.—Indicator Cards obtained by Dr. Watson.

seems preferable to a trembler coil, since over a very large range of current—in fact, whenever the current is large enough to cause the passage of a spark in the cylinder—the timing is exactly the same. The advantage of the trembler might be retained by using a switch, so that after the engine is started the trembler can be cut out, allowing the coil to act as a plain coil, a second condenser being provided.

The two diagrams shown in Fig. 106, obtained by Dr. Watson, illustrate the advantage, so far as economy is concerned, of advancing the spark more than usual when employing a very weak mixture—that is, when driving with the extra air valve as far open as possible. The lower figure is that obtained when the spark is as much advanced as is advisable when using a full mixture, and the i.h.p. at 1,000 revolutions was 2·36. In the upper figure the spark has been considerably further advanced, so as to allow for the slow burning of a weak mixture, and as a result the i.h.p. at 1,000 is 2·76, an increase of nearly 17 per cent. in power, the consumption of petrol remaining the same.

**133.** Mr. J. A. Davenport has also carried out some experiments at the East London College.* The only things allowed to vary were the high-tension spark gap and the **voltage of the cells.** One set of experiments was carried out at two volts and one at four volts in the cells, and the experimenter found to his surprise that the engine ran as well on two volts as four. In his own words, his conclusions were that "as far as the tests go, they show that for the coil used, the best spark gap is about 0·030 in. Further, since the current taken at two volts is half that taken at four volts, the cells will run four times as long in parallel as they will run in series; and that without injuriously affecting the consumption. Again, as the engine on a long run is slightly better at two than at four volts, everything is in favour of the use of two volts, instead of four volts as used in the standard practice." The engine used was a 4 h.p. one and the coil was of the non-trembler variety having a make-and-break on the half-speed shaft. Mr. Davenport's conclusions agree generally with Dr. Watson's as supporting the proposition that it is *the correct timing of the spark* rather than its "fatness" which is of importance. On the other hand, if conditions favour a "fat" spark probably one will take place even under circumstances which might prevent any "thin" spark passing at all.

## 134. Appendix.

### Description and Working of the Simms-Bosch High-Tension Magneto.

*Primary Winding.*—The end of the primary winding is connected to the brass plate 1. In the centre of this disc is screwed the fastening screw 2, which serves, in the first place, for holding the contact breaker in its place, and, in the second, for conducting the primary current to the platinum screw block 3, of the contact breaker. Screw 2 and screw block 3 are insulated from the contact-breaker disc 4, which is metallically connected with the armature core. The platinum screw 5 is arranged in the screw block 3. Pressed against this platinum screw by means of a spring 6 is the contact-breaker lever 7, which is connected to the armature core, and therefore with the beginning of the primary winding. The primary winding is

* *Engineering*, February 22, 1907.

therefore short circuited as long as lever 7 is in contact with platinum screw 5. The circuit is interrupted when the lever is rocked. A condenser 8 is connected in parallel with the gap thus formed.

*Secondary Winding.*—The beginning of the secondary winding is connected to the end of the primary, so that the latter is a direct continuation of the former. The end of the secondary winding leads to the slip ring 9, on which slides a carbon brush 10 which is

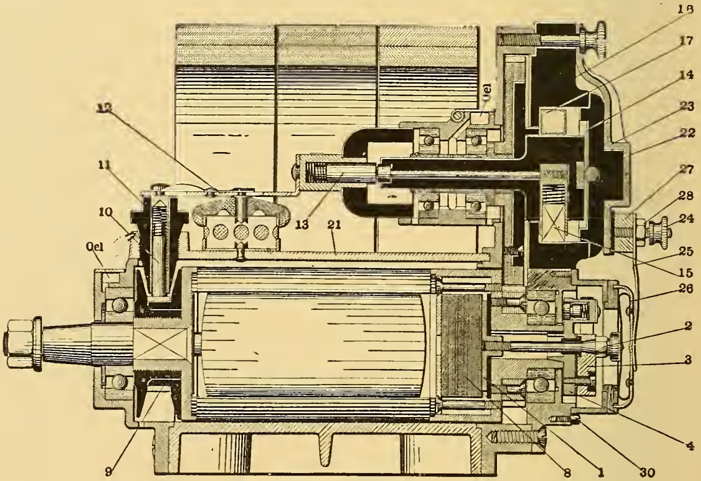



Fig. 107.—Arrangement of Bosch H.T. Magneto.

insulated from the magneto frame by means of the carbon holder 11. From the brush 10 the secondary current is conducted to the connecting bridge 12, fitted with a central carbon brush 13, and through the rotating distributor piece 14, which carries a radial contact carbon 15, to the distributor disc 16.

In the distributor disc 16 are embedded metal segments 17, of which there are three in the magneto of type "D 3," four in type "D 4" and six in type "D 6." During the rotation of the contact carbon 15, the latter makes contact with the respective segments, and always connects the secondary current with one of the contacts. Connected to the segments are sockets which serve for the reception of the contact plugs 18. These plugs serve as terminals for the cables leading to the sparking plugs of the individual cylinders.

From the end of the secondary winding, the high tension current is led to the respective cylinders, which are fixed alternately, then returns through the motor frame and armature core to the primary winding, and back to the beginning of the secondary winding.

*Method of Operation.*—The rotation of the armature in the magnetic field generates an alternating current in the armature winding, which twice attains its maximum during each revolution, the two maximums being 180° apart. An ignition may therefore be produced for each 180° rotation of the armature.

The tension of the current generated by the rotation of the armature is increased by short circuiting and opening the primary circuit through the contact breaker at the proper moment. At the moment the circuit is opened or interrupted, an arc light is formed at the sparking plug. As, however, the arc is only produced when the armature is in a certain position, which position must correspond to a definite position of the piston in the motor, it is necessary, that the armature of the magneto be driven positively from the motor.

*Speed of Rotation.* — The speed at which the magneto must be driven depends upon the number of cylinders.

In type "D 3," for instance, which is designed for three-cylinder motors, the armature must be run at a speed corresponding to three-quarters of the speed of the crankshaft. Type "D 4," for four-cylinder motors, must be run at the same speed as the motor, and "D 6," for six-cylinder motors, must be run at one and a half times the speed of the motor.

*Distribution of Current.*—The disc connected to the distributor brush 15 and which revolves the latter is geared in the different types from the armature shaft at such a ratio as to rotate the contact brush at the speed of the cam shaft of the motor.

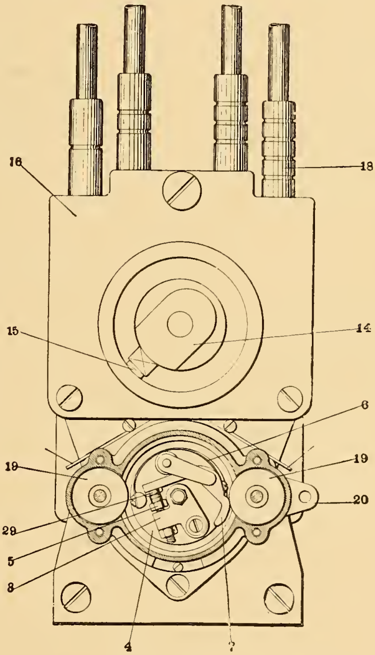


FIG. 108.

*Contact Breaker.*—The contact breaker is keyed to the armature shaft and is fastened by means of screw 2. It can be easily removed. Upon replacing the contact breaker, care should be taken that the key mentioned is placed in its key-way, and that screw 2 is well tightened up.

The short circuiting and interrupting of the primary circuit is effected twice during each revolution of the armature by means of the contact-breaker lever 7 on one hand, and the fibre rollers 19 on the other. As long as lever 7 is pressed against contact screw 5, the primary circuit is short circuited, the rocking of the lever through the fibre rollers 19 effect the break of the primary circuit, and at the same moment the ignition takes place. The movement of the lever 7 should not be more than ·5 mm. and may be adjusted by means of screw 5.

In order to protect the insulation of the armature and of the

current-carrying parts of the apparatus against excessive voltages, a safety spark gap is arranged on the dust cover 21. The current

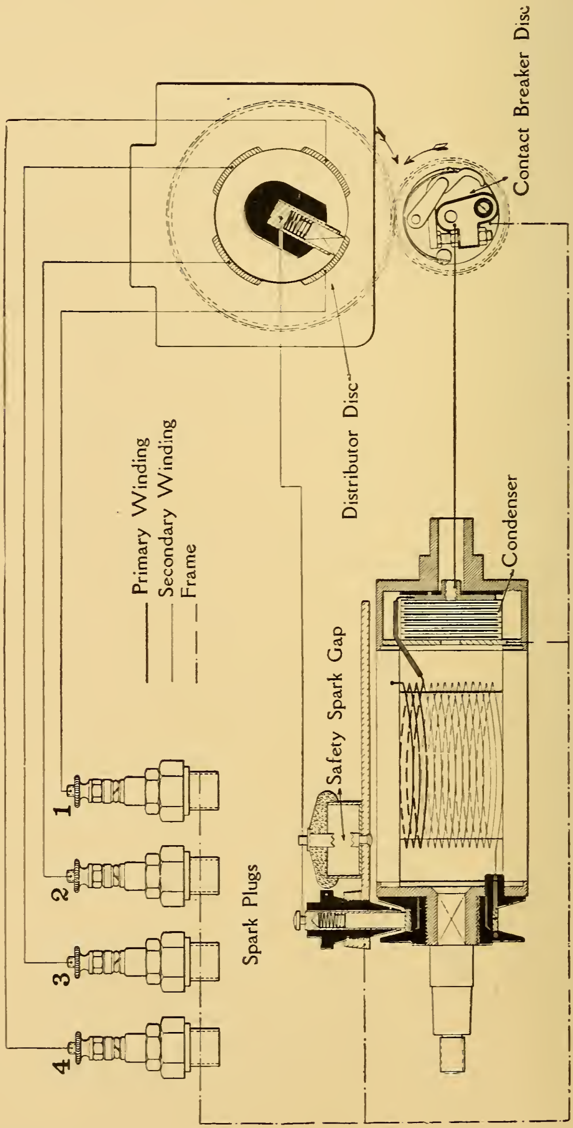



FIG. 109.—Wiring of Bosch H.T. Magneto and Circuit.

will pass through this gap in case a cable is taken off while the magneto is in operation, or if it should accidentally become dis-

connected. The discharges, however, should not pass through the safety gap for any length of time, especially not when the motor is equipped with a second system of ignition, and it is in such a case absolutely necessary to short-circuit the primary winding as above described, and thereby switch off the ignition.

### Timing of the Ignition.

The variation in the time of ignition is effected by causing the interruption of the primary circuit to take place earlier or later. For this purpose the timing lever 20 is arranged to be either advanced or retarded which produces either an earlier or a later interruption, and consequently an earlier or a later ignition. A variation of 40° on the armature spindle is thus possible, which is equal to over 50° on the shaft of the motor for three-cylinder motors, about 40° for four-cylinder motors and about 27° for six-cylinder motors.

## PROBLEMS.

1. Describe, with sketches, a gas or oil engine cylinder, showing valves and piston. (B. of E., 1907.)

2. Describe, with sketches, only *one* of the following—

A steam or gas engine governor, and how it regulates.

A spirit or oil engine for a motor car, showing how it drives the car and how it works.

(B. of E., 1901.)

3. Describe, with sketches, one, and only *one*, of the following—

Any form of governor.

An engine used on any kind of motor car.

4. Describe, with sketches, the carburettor of a petrol engine. (B. of E., 1907.)

5. The area of a petrol engine diagram is (using the planimeter which subtracts and adds properly) 4·12 square inches, and its length (parallel to the atmospheric line) is 3·85 in.; what is the average breadth of the figure? If 1 in. pressure represents 70 lb. per square inch, what is the average effective pressure? The piston is 3·5 in. in diameter with a stroke of 4 in. What is the work done in one cycle? If there are 800 cycles per minute, what is the horse-power?

*Ans.* 1·07 ins., 74·9 lb./in.$^2$, 240·3 ft. lb., and 5·82 h.p.

6. Describe, with sketches, the working of a good oil engine. How is the whole energy of the charge disposed of? (Mech. Sc. Tripos, Part I, 1898.)

7. Describe the method of balancing employed in any motor car engine known to you.

8. Describe carefully, with indicator diagram and sketches, the action of the oil in its passage through the oil engine. What is understood by (1) after-burning, (2) scavenging? What is the method adopted for governing the engine?

(Cambridge B.A. Degree, Old Regulations, 1904.)

9. Problem on carburettor design. Show how the calculations given in this chapter would be affected by the introduction of terms representing friction to passage of air and petrol. If, as is stated, the frictional resistance of petrol flowing in a pipe varies largely with temperature, show that definite calculation becomes far more difficult. (Experiment, however, would be simple, and students are recommended to undertake it as a piece of research work which would be important and valuable.)

## CHAPTER IX

# Petrol Engine Efficiency and Rating

Efficiency Tests under Various Conditions—Effect of Cylinder Dimensions on Efficiency—Engine Rating—R.A.C. Rule—Callendar Rule—Composition of Exhaust Gases as Related to Efficiency—Road and Air Resistance—"Gross-ton-miles-per-gallon" Measurement.

**135. Efficiency Tests on Petrol Motors.**—Among the most searching tests that have been carried out on petrol motors are those undertaken in the Engineering Laboratory at Cambridge under the supervision and guidance of Professor Hopkinson.

In one set of such tests * the engine used was a 16/20 h.p. Daimler four-cylinder engine capable of running at 250 to 1,400 revs. per min. Other particulars were—

| | |
|---|---|
| Total volume of one cylinder with piston on out centre . . . | 0·04 cu. ft. |
| Volume of compression space . | 0·0104 cu. ft. |
| Compression ratio . . . . | 3·85 |
| Diameter of cylinder . . . . | 3·56 inches = 90 mm. |
| Length of stroke . . . . . | 5·11 inches = 130 mm. |

The type of indicator used was one invented by Professor Hopkinson. It was of the piston type, the piston being forced against the mid point of a piece of spring steel held at both ends, the deflection of which rotated a mirror through an angle, and so moved a spot of light on a screen. The mirror mechanism was rocked at the same time in a perpendicular direction, corresponding to the motion of the piston. These two motions combined to give the usual

* *Engineering*, February 8, 1907.

indicator diagram, which was then thrown on to a screen or photographed. This is an adaptation of a principle first used by Professor Perry many years ago.

The tests involved three sets of measurements—(1) engine losses, (2) b.h.p., and (3) fuel consumption. From (1) and (2) the i.h.p. could be obtained, and therefore the mechanical efficiency. The tests were run with the carburettor as fitted by the engine builders, and it must not therefore be taken that the engine was of necessity adjusted to give maximum power or efficiency.

The results of the tests are shown in Fig. 110 in which curves are given for the i.h.p., b.h.p., the mean effective pressure and the torque on the crankshaft. It will be seen that the m.e.p. is nearly constant and equal to about 85 lb./in$^2$. The mechanical efficiency varies from 85 to 75 per cent.—falling slowly as the speed exceeds 600 revs. per min. The petrol used had a thermal efficiency on the lower scale of 17,000 B.T.U. per lb. and on this basis the following table of thermal efficiencies was calculated—

| Speed. | Petrol Consumption (Pounds). | | | Thermal Efficiency. | |
|---|---|---|---|---|---|
| Revs. per Minute. | Per I.H.P. Hour. | Per B.H.P. Hour. | Per 1,000 Revs. | On I.H.P. | On B.H.P. |
| 400 | 0·78 | 0·9 | 0·30 | 18·6 | 16·1 |
| 400 | 0·75 | 0·87 | 0·28 | 19·3 | 16·6 |
| 600 | 0·685 | 0·81 | 0·26 | 21 | 17·9 |
| 600 | 0·655 | 0·77 | 0·24 | 22 | 18·8 |
| 800 | — | — | 0·24 | — | — |
| 1,000 | 0·6 | 0·75 | 0·22 | 24·2 | 19·3 |
| 1,000 | 0·6 | 0·75 | 0·206 | 24·2 | 19·3 |
| 1,100 | 0·59 | 0·785 | 0·202 | 24·6 | 18·4 |
| 1,225 | (0·65) | 0·94 | 0·22 | (22·3) | 15·4 |

NOTE.—At speeds 400, 600, and 1,000, two tests are given to show the range of variation. At 1,225 the indicated horse-power is uncertain, as no direct measurement of loss was made at that speed.

The thermal efficiency rises considerably with increase of speed—due no doubt in part to there being less time for the

cold walls to cool the explosive mixture, but in view of the variability of the composition of mixture passed by the carburettor (of the usual jet type) it is not safe to build any deductions on these measurements, although it is very useful to have them recorded. An interesting measurement in addition to the above was that of the pressure in the induction pipe. With the throttle wide open and a speed of

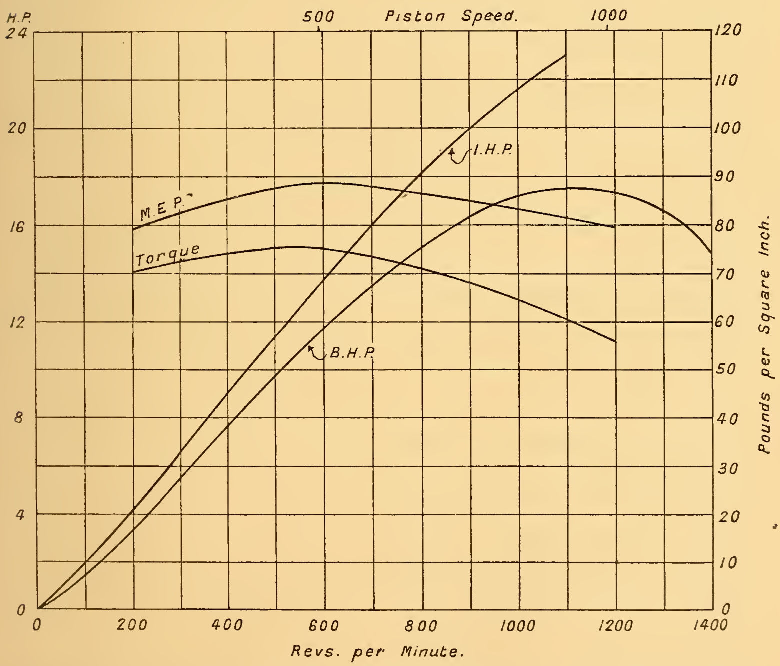



FIG. 110.—Professor Hopkinson's tests on 16/20 H.P. Daimler Engine.

1,000 r.p.m. this pressure was about $1\frac{1}{2}$ lb./in$^2$. below atmospheric pressure. With the speed reduced to 400 r.p.m. this pressure was less than $\frac{1}{2}$ lb./in$^2$. below atmosphere, showing that either the carburettor was supplying a richer mixture at the lower speeds, or that only $\frac{2}{3}$ as much air is taken per revolution at 1,000 as at 400 revs. The design of the average carburettor provides for over-supply of extra air at high speeds rather than under-supply, and the mixture probably was richer at 400 than 1,000 revs.

Both the causes suggested were therefore in all probability at work.

**136.** Some tests * to discover the relation between the rate of consumption of petrol per b.h.p., and **the character of the spark** employed were undertaken at the Central Technical College by Messrs. Topham and Tisdall. A small single cylinder De Dion engine was used of 66 mm. bore, and 70 mm. stroke. The proportion of air to petrol varied with atmospheric conditions and was adjusted for each test so as to give a maximum power development, the throttle was kept fully open, and the air supply alone was varied. The engine was run at full normal speed. The chief results obtained were as follows—

| Source of Spark (high tension). | Petrol Consumption in galls per B.H.P. Hour. |
|---|---|
| Accumulator with no external gap . | 0·208 to 0·187 |
| Magneto with external gap . . . | 0·289 to 0·249 |
| Accumulator with external gap . | 0·226 to 0·24 |

Not much can be gained from these measurements, though it would seem that different forms of high-tension ignition have a generally similar effect on the rate of fuel consumption, the accumulator having a slight advantage. The result of using an external spark gap in the high-tension circuit is worth recording. The plug insulation resistance was over 1,000 megohms cold, but after being used for running the engine for five minutes, it fell to about 6 megohms. At a dull red the resistance fell further to 2 megohms, and eventually sparking ceased entirely. On the introduction of an external air gap in the circuit, however, sparking was resumed, even when the plug was made bright red hot, and the resistance had sunk as low as 800,000 ohms. The experimenters considered that this showed that leakage through the porcelain of the plug might become quite large enough to stop sparking entirely, and that the effect of the introduction of the external air gap was to prevent the application of the

* *Engineering*, December 28, 1906.

voltage across the sparking points until the instant at which current began to flow in the high-tension circuit, the spark then being of the nature of a condenser discharge, and the leakage being considerably reduced in quantity.

**137. The Effect of Cylinder Dimensions on Efficiency.**—A good many attempts have been made to produce a working theory of cylinder dimensions, particularly of cylinder diameters, as affecting the economy and power of internal combustion engines. Most of the attempts have owed their origin to the competitive trials of motor cars in which the various vehicles are classed according to their horse-power and which therefore require that the figures given should be properly comparable. If no such precautions are taken, it becomes possible, for instance, for the owner of a high powered car to enter it as of small h.p., and so give it an unfair advantage, say in hill-climbing, over a car the power of which was really this figure. For large internal combustion engines the existence of such a rule is not of vital importance, since the comparison of one engine and another depends upon so many other factors, and moreover organized competitions are unknown.

This subject has been discussed by **Professor Callendar,** in a paper before the Society of Automobile Engineers * under the title of "The Effect of Size on the Thermal Efficiency of Motors." Cylinders were considered of as great diameter as 14 inches which, although not very large for a gas engine, was well outside the range of motor car cylinders. This makes the paper the more valuable in the general sense, even if from the strictly motor car point of view it may appear that advantage would have been gained had it been possible to build up the theory upon engine trials with cylinders of more nearly the customary motor car size. As it is, however, the paper presents a general theory not only applicable to motor cars but to larger engines also and few writers on the subject have exhibited so skilful a scientific treatment, or so able a grasp of the essential problems presented. For details, reference must of course be made to the actual paper, but a brief

* May 8, 1907.

description and discussion may with advantage be given here.

**138.** It is well known, and it has been discussed in an earlier chapter, that the "air standard" of efficiency is much higher than the efficiencies obtained in practice, and that the ratio of the latter to the former is commonly about 60 per cent. This means a deficit of 40 per cent. owing to some cause or other. What is this cause? The answer is, first, that the "air standard" of efficiency is a far higher one than any actual engine can ever achieve, owing to the fact that whereas the value of $\gamma$ assumed in the "air standard" equation is 1·40, its average value for the actual cylinder gases at working temperatures, taking the increase of specific heat into account, would be more nearly 1·3. This alone accounts for about 20 per cent. of the 40 per cent. apparently lost, and the remaining 20 per cent. is due to various heat losses such as jacket loss, exhaust loss, radiation loss, etc. In general, therefore, it would appear that the 40 per cent. loss is divided about equally between the two, but a more exact inquiry into the matter has shown Professor Callendar that it would be more correct to put the unavoidable apparent loss due to the properties of the gases down as 25 per cent., and the remaining 15 per cent. to the loss of efficiency owing to heat losses during the operations of the cycle. It is quite clear that, as the volume of gas in a cylinder will be proportional to the cube of the dimensions, and the surface of the cooling walls proportional only to the square of the dimensions, doubling the size of an engine will halve the heat losses due to cooling. The fact that the larger engine will probably not run at so high a speed has very little effect on this conclusion, as although the time the gases will have to cool will increase with diminishing speed, yet the diminished speed will lead to diminished scrubbing of the cylinder walls by the molecules of the gases, and so leave matters much where they were. It may therefore be said that the loss of efficiency due to cooling will be $\propto \frac{1}{D}$ where $D$ is the cylinder diameter. Some losses, however, such as the exhaust

loss, are not due to cooling, and what law of dimensions they will follow it is not easy *à priori* to say. Still, **the most important loss** has been shown to be proportional to $\frac{1}{D}$, and as an attempt at a working theory, there is no harm in grouping the losses together and putting them proportional to $\frac{1}{D}$. This is what Professor Callendar does with several sets of engine trials. One set is based on his own experiments on an engine with a cylinder diameter of 2·36 inches, and the others are drawn from the Report of a Committee of the Institution of Civil Engineers. From the combined results on all four engines, he finds that the value of the constant $a$ to be put in the expression—

$$\text{loss of efficiency} = \frac{a}{D}$$

actually comes out as 1·0 when $D$ is measured in inches. So that loss of efficiency $= \frac{1}{D}$. This enables the following table to be drawn up.

| | | | | |
|---|---|---|---|---|
| Designation of engine . . . . | $C$ | $L$ | $R$ | $X$ |
| Diameter of cylinder, inches . . | 2·36 | 5·5 | 9·0 | 14·0 |
| Loss of efficiency $\left(=\frac{1}{D}\right)$ . . | 0·42 | 0·18 | 0·11 | 0·07 |
| Resulting efficiency figure $\left(1-\frac{1}{D}\right)$ | 0·58 | 0·82 | 0·89 | 0·93 |
| Observed relative efficiency as compared with air standard . | 0·44 | 0·61 | 0·65 | 0·69 |
| Ratio of last two lines . . . . | 0·76 | 0·75 | 0·73 | 0·74 |

**139.** The author understands that the value of the above constant $a$, viz. 1·0, was obtained by Professor Callendar mainly, if not entirely, from the above table. It was obviously necessary in accordance with the theory that the figures in the bottom line should come out the same, or as nearly the same as possible, and the value of $a$ which was found most nearly to do this, *happened* to be 1·0 ; had the

cylinder diameters been measured in millimetres the value of $a$ would have been 25·4.

In this way the relative efficiency of any engine can be written down as $0{\cdot}75\left(1-\frac{1}{D}\right)$ and if the "air standard" efficiency for the degree of compression under consideration be called $E$, then the

$$\text{thermal efficiency} = 0{\cdot}75E\left(1-\frac{1}{D}\right)$$

Had the "air standard" been a standard really applicable directly to gas engines, the figure 0·75 would have been unity, so further simplifying the formula. As it is the above equation shows that even the largest engines cannot get nearer the "air standard" than 75 per cent. The establishment, even provisionally, of such a rule as this has an important influence in the consideration of the effect of dimensions on engine performance.

**140.** The Royal Automobile Club published, in 1906, the following **engine rating** based on cylinder diameter :

Nominal b.h.p. $= 0{\cdot}40D^2$ per cylinder, where $D =$ cylinder diameter in inches.

It has been found to give results which follow very nearly the ratings in general use in the motor world. It is equivalent to assuming that the piston speed is constant and equal to 1,000 feet per minute, and that the mean effective pressure on the piston during explosion is 67·2 lb. per square inch. It takes no account of changes in mean pressure owing to change in dimensions or in the degree of compression, nor for any change in piston speed owing to alterations in the ratio of stroke to bore or for other reasons. Professor Callendar points out that by altering the compression ratio from the 3·5 customary with petrol engines to 5·0 about 20 per cent. more power could be obtained than the R.A.C. rating would give.

The R.A.C. rating depends on the fact that with fixed piston speed and a constant value for the mean pressure the h.p. will vary with the area of the piston, i.e. vary as $D^2$. But this assumes that all sizes of engines are equally efficient

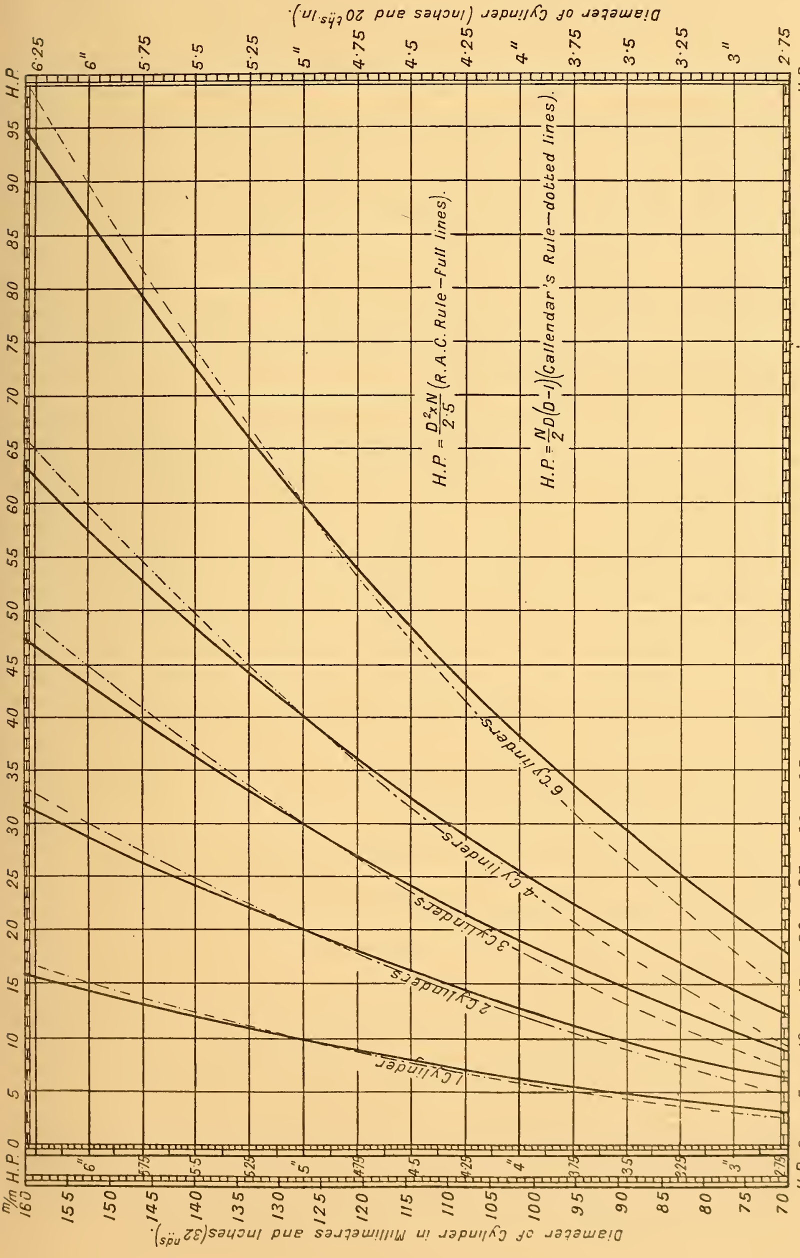



FIG. 111.—"R.A.C. Rating" for comparing the H.P. of Petrol Engines. Prof. Callendar's rule is also given.

when worked under the same conditions, which is not the case. If $D^2$ be multiplied by the expression $\left(1-\frac{1}{D}\right)$ as proportional to the efficiency, the result is to obtain the expression $D(D-1)$ which therefore should be used in place of $D^2$ in the rating formula. In this way Professor Callendar suggests a "P.C. rating" (Petrol Consumption rating) of

$$\text{b.h.p.} = \frac{D}{2}(D-1)$$

the figure 2 being the suitable constant. The P.C. ratings and R.A.C. ratings for a number of cylinder diameters are given in the following table—

| *D.* | R.A.C. Rating (per cylinder). | P.C. Rating (per cylinder). |
|---|---|---|
| Inches. | B.H.P. | B.H.P. |
| 1 | 0·40 | nil |
| 2 | 1·6 | 1·0 |
| 3 | 3·6 | 3·0 |
| 4 | 6·4 | 6·0 |
| 5 | 10·0 | 10·0 |
| 6 | 14·4 | 15·0 |
| 8 | 25·6 | 28·0 |
| 10 | 40·0 | 45·0 |
| 20 | 160 | 190 |

These two formulae are shown plotted in Fig. 111. The Callendar formula $\frac{D}{2}(D-1)$ gives the h.p. per cylinder; putting it in the same form as the R.A.C. formula it would be $\frac{DN}{2}(D-1)$.

The result of using the P.C. rating would be, to quote Professor Callendar :—"According to the R.A.C. formula, a four-cylinder engine with 2 inch bore and stroke (like the F.N. motor cycle engine), is rated at 6·4 h.p., and is equivalent to a single-cylinder of 4 inches bore. According to my experiments the four-cylinder of 2 inch bore could not develop much more than 4 h.p. under ordinary conditions, and would stand no chance against the single-cylinder of 4

inches bore. A four-cylinder of 3 ins. bore is equivalent to a single-cylinder of 6 in. bore by the A.C. rating, but according to the P.C. rating, the single-cylinder would have an advantage in point of power of 25 per cent. A two-cylinder of equal power on the A.C. rating would have an advantage of about 12 per cent. over the four-cylinder, and a six-cylinder a disadvantage of about 10 per cent." Professor Callendar also remarks :—"An obvious objection to the P.C. type of formula is that the b.h.p. of an engine of 1 in. bore and stroke would be zero. According to the R.A.C. rating it should be $\frac{2}{5}$ h.p. It would no doubt be possible to get such an engine to run if very delicately made, but the effect of ignition lag would be serious at the normal speed of 6,000 revolutions per minute, and I doubt whether it could be made to give as much as $\frac{1}{10}$ h.p. on the brake."

**141.** It may be remarked incidentally that small cylinders have a considerable advantage in h.p. per lb. of weight, as while the h.p. only increases according to the square of the piston diameter, the weight varies more nearly as the cube, and if made to the same drawings, but to different scales, would be actually as the cube.

The P.C. rating of $\frac{D}{2}D(—1)$ requires to be corrected for the possible variation of compression ratio, and this can be done by multiplying by an expression proportional to the "air standard" efficiency, and using a suitable constant. So corrected, the P.C. formula becomes :

$$\text{b.h.p.} = 2{\cdot}5E\frac{D}{2}(D-1)$$

$$\text{or } 1{\cdot}25ED(D-1).$$

This formula should therefore be used when it is desired to compare several engines of different sizes on the basis of their petrol consumption. Professor Callendar points out that according to his investigations the effect of increasing cylinder diameter from 2 in. to 4 in. is equivalent in efficiency improvement to increasing the compression ratio from 3 to 5. It would be very interesting to test this experimentally.

One would have anticipated that such an increase in compression ratio would have had much the larger effect.

Professor Callendar also deals with the modification that would be required to be made in his P.C. formulae if the ratio of stroke to bore became much more than unity, and he gives the modified formula as

$$\frac{D}{2}(D-1)+\frac{xD}{4L}$$

Where $L$=stroke and $x=L-D$.

The basis of this correction is however somewhat uncertain, and as the correction itself becomes a very small one in practice, it need not usually be taken into account. Investigation is also made into the effect of a change in piston speed to which a change in the ratio of stroke to bore might naturally be expected to lead, and the following expression for the piston speed is suggested—

$$\text{Piston speed}=1{,}000\left(1+\frac{x}{2D+L}\right)\text{ feet per min.}$$

Such a formula gives a piston speed varying from 1,000 to 1,100 as the ratio of stroke to bore increases from 1/1 to 4/3. In the case of *hill climbing* contests, it is desirable that a correction should be made so as to enable the true b.h.p. to be obtained from the cylinder dimensions and all that varies therewith. To this end, Professor Callendar establishes what he calls an H.C. rating in this form—

$$\text{H.C. rating b.h.p.}=\frac{D}{2}(D-1)+\frac{xD}{6}$$

This of course may also be multiplied by a term proportional to $E$ to correct for compression ratio variation.

**142.** An interesting point is to find out what is **the best compression ratio** for minimum weight of engine per b.h.p. developed.

If the weight $\propto D^3$ (though in many engines $D^{2\cdot5}$ would be nearer the mark), and the h.p. $\propto D(D-1)$

Then

$$\frac{\text{h.p.}}{\text{weight}}=\frac{D(D-1)}{D^3}=\frac{D-1}{D^2}\text{ which}$$

gives the following table :—

| | | | | | | |
|---|---|---|---|---|---|---|
| $D$ :— | 1 | 2 | 3 | 4 | 5 | 10 |
| $\dfrac{D-1}{D^2}$ :— | 0 | 0·25 | 0·22 | 0·19 | 0·16 | 0·09 |

showing that the greatest h.p. per lb. weight of motor would be obtained when $D=2$ inches, although $D=3$ inches gives almost as good a result, and in cases where the weight is proportional to $D^{2\cdot5}$ the value $D=3$ gives the best result. The weight will however be affected by any change in *the compression ratio* since with increasing compression the engine parts must be made heavier. Professor Callendar makes an estimate of the effect of compression ratio thus :—" If we assume that the maximum pressure is nearly proportional to the compression pressure, and that the strength and weight of the motor and its gear must be proportional to the maximum stress, we may take the compression pressure as a measure of the weight. Taking the air-standard efficiency $E$ as a measure of the mean pressure and the power, we find that the weight ÷ power ratio is nearly twice as great for a compression ratio 5 as for a compression ratio 2, and diminishes with diminution of compression-ratio, reaching a minimum at $r=1\cdot9$ if $\gamma=1\cdot40$. The best value of the compression ratio depends on various conditions, but chiefly on the ratio of the load carried to the weight of the motor and its accessory gear. If the load is equal to the weight of the motor, the compression ratio should be rather less than 3. If the load is double the weight of the motor, the compression should be about 4·5. The above shows that in many cases where weight-saving is a primary consideration it may be desirable to reduce the compression considerably, and that even in automobiles there is no great advantage in increasing the compression ratio beyond 4·0, provided that lightness of construction and smoothness of running are duly considered in the design of the motor." These conclusions are of very great interest, though owing to the complexity of the phenomena it is not to be expected that they will command by any means universal assent.

**143.** *Exercise.*—The great desideratum for high efficiency

is large cylinder capacity in proportion to the internal cooling area of the cylinder. This may not make for "flexibility," as that is increased by the existence of "pockets" in the cylinder end, and "pockets" mean a considerable addition to cooling area. What ratio of cylinder diameter to length makes the ratio

$$\frac{\text{area of exposed surface inside cylinder in sq. ins.}}{\text{capacity of cylinder in cubic inches}}$$

a minimum? The answer to this question is the answer we want.

Let the volume enclosed be $V$, and cylinder diameter be $2a$.

Then $$\text{height} = \frac{V}{\pi a^2} = h$$

The internal surface of cylinder (including the surface of piston face)

$$= 2\pi a^2 + 2\pi a h$$

Therefore $$\text{Ratio } \frac{\text{surface}}{\text{content}} = \frac{2\pi(a^2 + ah)}{V}$$

Substitute for $h$ as above

and $$\text{Ratio} = \frac{2\pi}{V}\left(a^2 + \frac{V}{\pi a}\right)$$

To find when this is a minimum differentiate and equate to zero.

Therefore $$2a - \frac{V}{\pi a^2} = 0.$$

and $$2a = \frac{V}{\pi a^2} = h.$$

Therefore diameter and height must be equal to produce the desired effect or the ratio of stroke to bore must be unity. This interesting result bears out actual practice inasmuch as in most engines the ratio of stroke to bore is always nearly unity.

Suppose however that "pockets" are introduced, so that the surface is increased whilst the capacity is but slightly affected, what effect does this produce on the above calculation?

Surface now $=2\pi a^2+2\pi ah+ca^2$ where $c$ is some constant

$$=(2\pi+c)a^2+2\pi ah$$

as before $\quad h=\frac{V}{\pi a^2}$ very nearly

so that $\quad \text{Ratio } \frac{\text{surface}}{\text{volume}}=\frac{(2\pi+c)a^2+2\pi ah}{V}$

$$=\left\{\frac{1}{V}(2\pi+c)a^2+\frac{2V}{a}\right\}.$$

Differentiate and equate to zero.

Therefore $\quad 2(2\pi+c)a-\frac{2V}{a^2}=0.$

or $\quad (2\pi+c)a=\frac{V}{a^2}=\pi h.$

So that $\quad \text{Ratio } \frac{h}{2a}=\frac{2\pi+c}{2\pi}=1+\frac{c}{2\pi}$

so that ratio of stroke to bore will be greater than unity. $\frac{c}{\pi}$ is of course the fraction of the piston area of which the pocket increases the internal area of the cylinder. If this ratio $=\frac{1}{4}$ then $\frac{h}{2a}=1\cdot125$, giving a relation which is quite close to that adopted in practice with engines which have "pockets."

**144. Composition of Exhaust Gases.**—The efficiency of a petrol engine naturally depends on the degree to which combustion is complete. The exhaust gases should not, for good efficiency, contain any CO. All the carbon present should be burnt to $CO_2$. Nor, if the proportion of air is closely adjusted, will there be any oxygen in the exhaust.

It is difficult to write down the chemical formula in accordance with which the combustion of petrol takes place in an atmosphere of air, owing to the complex nature of the petrol molecule. An example of a chemical formula for a sample of petrol is

$$41\cdot86C_6H_{14}+6\cdot48C_7H_{16}+C_5H_{12}.$$

It is not easy to work equations containing a substance like

that. But since $C_6H_{14}$ (or hexane) is by far the largest constituent it is interesting to write down the combustion equation for hexane, and to look upon it as representing, roughly, what occurs with petrol. In any case no exact theory for petrol could be worked out as the composition varies so greatly between the different qualities used in practice.

$C_6H_{14}$ burns with $O_2$ as follows—

$$2C_6H_{14}+19O_2=12CO_2+14H_2O$$

so that 21 volumes of mixture give 26 volumes of products, or, if the steam be condensed to water, 12 volumes.

In actual working, ordinary air and not pure oxygen is used, so that there is nitrogen also to be considered. With 19 volumes of oxygen, $19 \times 3{\cdot}76$, or 71·3 volumes of nitrogen would be associated—making a total of 90·3 volumes of air. Each volume of hexane therefore requires 45·1 volumes of air for complete combustion and the equation can be therefore rewritten as—

$$2C_6H_{14}+19O_2+71{\cdot}3N_2=12CO_2+14H_2O+71{\cdot}3N_2.$$

The right-hand side of this equation is exhaust products, and the composition by volume will be—if the volume of the water be neglected—$\frac{12}{83{\cdot}3}$ or **14·4 per cent. of $CO_2$ and 85·6 per cent. of $N_2$**. If too little air were admitted some of the $CO_2$ would be reduced to CO, which being a poisonous gas is a very undesirable element in the exhaust ; moreover, it reduces the thermal value of the gas owing to a part of the carbon not being completely oxidized—a loss which has already been dealt with quantitatively in the chapter on suction producer gas. If too much air is admitted, free oxygen will appear in the exhaust. We have therefore the following three deductions :—

(1) When oxygen occurs in the exhaust too much air has been admitted

(2) Too little air leads to formation of the poisonous CO.

(3) When neither CO nor $O_2$ appear in the exhaust the air is in just the right proportion.

It will now be the easier to understand some experiments

on the composition of exhaust gases made by Professor Bertram Hopkinson and Mr. L. G. Morse. The experiments were made in the engineering laboratories at Cambridge, on the Daimler engine already referred to.

The speed was kept at 700/750 r.p.m., and a jet carburettor of the usual sort was used. The throttle was kept open so that the suction never exceeded ½ lb. per sq. inch in the inlet pipe close to the inlet valves. Fuel used was Pratt's motor spirit; density 0·715 to 0·720; Calorific value 18,900 B.T.U. (lower value). The exhaust gases were analysed by the ordinary volumetric methods, the $CO_2$ being absorbed by potash, the oxygen by pyrogallol, the CO by an acid solution of cuprous chloride, and the $H_2$ by palladianized asbestos.

The following table shows the results recorded—

Experiments made by Professor Bertram Hopkinson and Mr. L. G. Morse.

| | | | | | | |
|---|---|---|---|---|---|---|
| Petrol consumption in lb. per 1,000 revs. | 0·181 | 0·191 | 0·197 | 0·217 | 0·250 | 0·293 |
| Brake load at 43 in. radius . . . . lb. | 25 | 27·5 | 29·3 | 29·4 | 29·3 | 27 |
| Thermal efficiency . . | 0·244 | 0·252 | 0·261 | 0·238 | 0·204 | 0·162 |
| $CO_2$—measured . . . | 10·9 | 12·8 | 13·5 | 10·6 | 9·6 | 6 |
| $O_2$ ,, . . . . | 3·6 | 1·5 | 0·2 | — | — | — |
| CO ,, . . . . | — | — | 0·7 | 5 | 6·25 | 11·6 |
| $H_2$ ,, . . . . | — | — | — | 2·1 | 2·65 | 8·7 |
| $N_2$ by difference . . | 84 | 84 | 84 | 81 | 80 | 73 |
| Total $O_2$, calculated from $N_2$ . . . . | 22·4 | 22·4 | 22·4 | 21·5 | 21·3 | 19·4 |
| $H_2O$ calculated . . . | 15·8 | 16·2 | 16·8 | 16·8 | 17·2 | 15·2 |

**145.** From this it will be seen that when the CO and $O_2$ are at a minimum, the $CO_2$ is 13·5 per cent. and the $N_2$ 84 per cent., **figures which are very close** to those calculated above from the chemical formula. Moreover, it will be seen that it is at this point that the highest thermal efficiency (0·261) was recorded. This is best brought out when the points are plotted in a curve as in Fig. 112.

Curve *B* (thermal efficiency) shows how quickly the thermal efficiency declines when CO begins to be produced. Curve *A* (corresponding to b.h.p.) is of a very different form, as although it is true that minimum production

of CO corresponds to an output in h.p. very little less than the maximum, yet that maximum is found when the CO amounts to 0·7 per cent. and is very nearly maintained even when the proportion of CO rises to over 6 per cent. The ideal condition of working is obviously the apex of Curve *B*, but the corresponding point on Curve *A* is not a convenient one to work at. In all engineering work it is customary to work, if possible, near to the middle of a curve which has a slow hump such as Curve *A* in order that small variations to right or left may not make much difference. The ideal working point on Curve *A* is therefore by no means the most convenient practical point, which in this case would correspond to about 5 per cent. of CO. This temptation has led engine builders to set carburettors so to give **maximum power,** instead of aiming at minimum production of CO and therefore **maximum thermal efficiency.**

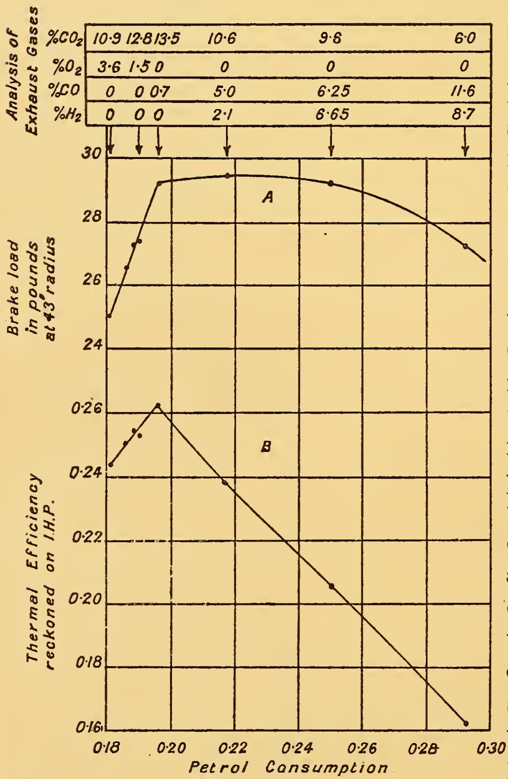


FIG. 112.

Mr. Dugald Clerk has also carried out tests of this kind. He used for this purpose the engine on his 18 h.p. Siddeley car. The engine was a 4-cylinder one, bore 4 inches, stroke 4 inches. Samples of the exhaust were taken while—

The following table shows the composition of the exhaust gases under these conditions on three different occasions—

EXPERIMENTS BY MR. DUGALD CLERK ON EXHAUST GASES FROM 18 H.P. SIDDELEY CAR. EXHAUST GAS ANALYSIS BY MR. BALLANTYNE.

| | (*a*) Car standing; engine running as slowly as possible. No load. | | | (*b*) Car standing; engine running at about 600 revs. per min. No load. | | | (*c*) Car running on level about 18 miles per hour. Throttle less than half open. | | | (*d*) Car climbing a hill; engine running about 1,000 revs. per min. Throttle full open or over three-quarters open. | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Apr. 23 | May 7 | July 3 | Apr. 23 | May 7 | July 3 | Apr. 23 | May 7 | July 3 | Apr. 23 | May 7 | July 3 |
| | * | * | | * | | | | | | | | |
| Carbonic Oxide, $CO$ | 0·5 | 0·4 | — | none | 3·6 | 1·8 | 6·9 | 4·2 | 2·4 | 3·6 | 3·6 | 2·2 |
| Hydrogen, $H$ | 0·2 | 0·1 | — | none | 1·2 | 0·6 | 2·4 | 1·4 | 0·8 | 1·2 | 1·3 | 0·8 |
| Methane, $CH_4$ | none | none | — | none | 0·3 | 0·2 | 0·9 | 0·5 | 0·3 | 0·4 | 0·3 | 0·3 |
| Hydrocarbon vapours | trace | trace | — | trace | trace | trace | trace | trace | trace | trace | trace | trace |
| Carbonic acid, $CO_2$ | 5·4 | 6·0 | — | 6·3 | 10·8 | 5·6 | 9·9 | 11·5 | 11·0 | 11·7 | 12·2 | 11·8 |
| Oxygen, $O$ | 11·8 | 10·2 | — | 11·2 | 2·2 | 10·6 | 0·3 | none | 2·2 | 0·1 | none | 0·6 |
| Nitrogen, $N$ | 82·1 | 83·3 | — | 82·5 | 81·9 | 81·2 | 79·6 | 82·4 | 83·3 | 83·0 | 82·6 | 84·3 |
| Totals | 100·0 | 100·0 | — | 100·0 | 100·0 | 100·0 | 100·0 | 100·0 | 100·0 | 100·0 | 100·0 | 100·0 |

* Engine had been running for some hours. Engine in other cases had been just started from all cold.

(*a*) The car was standing on the level with the engine running as slowly as possible.

(*b*) The car still standing, but engine running at about 600 r.p.m.

(*c*) The car running on a level at about 18 m.p.h., the throttle less than half open.

(*d*) The car climbing a hill, engine running about 1,000 r.p.m., and throttle from three-quarters to full open.

It will be seen from the above table that under circumstances which might quite often occur in practice about 4 per cent. of CO is being produced. To meet the difficulties of designing a carburettor which should mix air and petrol in constant proportions under all conditions of load and speed is no easy thing, indeed most builders aim at quite different mixtures viz., those that make for ease at starting, for rapidity of "picking up," and other features of car management that make for ease of manipulation. Mr. Dugald Clerk is so sensible of its difficulty that he has not hesitated to make the suggestion contained in the following extract from his paper—

"As the problem is to charge any given volume of air passing into the engine with a practically unvarying proportion of petrol vapour in an uniform manner, it seems to me that all systems of speed control must fail to obtain proportionality throughout the whole range. If the entering air could be made to drive a fan, similar say to an anemometer, with proper precautions this fan or anemometer could be made to run at a speed of rotation very closely proportional to the volume of air supply throughout the whole range of conditions. If a spindle be driven by this anemometer, and a small chain passed from that spindle into a petrol vessel at constant level, then it would be possible to work out a carburetting contrivance which would supply the air passed through it on its way to the engine with petrol exactly in proportion to the charge volume taken in. I believe contrivances of this kind have been proposed, and it seems to me that although we have practical difficulties, if correctly carried out, they should be able to give a perfectly uniform mixture

to an engine under all conditions of speed variation and charge volume variation."

Mr. Dugald Clerk concludes from the results of his experiments that the following conditions appear to produce imperfect combustion—

1. "Too rich mixture with insufficiency of oxygen.
2. Too weak mixture with excess of oxygen, but too slow a rate of ignition and combustion.
3. Irregular mixture—mixture supplied too rich in composition at one part of the stroke, and too weak in another; that is, bad mixture.
4. Engine and carburettor cold. This tends to cause imperfect combustion, due partly to low temperature and partly to bad carburetting.
5. Improper timing of ignition, and missed ignitions.
6. Igniting in the body of the cylinder, instead of in a port. This will produce imperfect combustion at light loads."

The sixth of the above conditions is an exceptionally interesting one. It seems incontestably to be the case that the presence of "pockets" in cylinders leads to loss of efficiency, "pockets" being the name given to any recess in the top of a cylinder which has the effect of increasing the clearance volume. But it is almost equally certain that the presence of "pockets" improves the running of the engine under working conditions: and renders it, as it is termed, "more flexible." Pockets lower the efficiency as they increase the ratio of surface to volume, but they render ignition more certain, as, even at light loads when the exhaust products left in the clearance space dilute the incoming charge very considerably, there is the likelihood of the neighbourhood of the sparking plug (which is probably situated in or near one of these pockets) being rich locally in explosive mixture, so ensuring the proper starting and timing of the ignition. For this reason pockets are popular.

**146. Combustion of Paraffin.**—Average paraffin consists chiefly of decane of which the chemical symbol is $C_{10}H_{22}$. The combustion of decane in air may be represented chemically by the formula

$$2C_{10}H_{22}+31O_2+116{\cdot}5N_2=20CO_2+22H_2O+116{\cdot}5N_2$$

The composition by volume of the exhaust gases when cooled to normal temperature and pressure will therefore be $\frac{20}{136{\cdot}5}$ or 14·6 per cent. of $CO_2$ and 85·4 per cent. of nitrogen. If 10 per cent. less air were admitted the combustion would be incomplete, as on the average 6·2 atoms of carbon would get oxidized to the extent of CO only. This would lead to there being $\frac{6{\cdot}2}{104{\cdot}9+6{\cdot}2+13{\cdot}8}$ or 5 per cent. by volume of CO present in the exhaust gases. These exhaust gases are not identically the same as those produced in the working of an internal combustion engine which uses paraffin as its fuel for two reasons—

(*a*) Paraffin is not pure decane.

(*b*) The gaseous mixture which is ignited in the cylinder consists in part of some of the exhaust products of the previous explosions and the composition must depend a good deal on the history of the explosions of the last few strokes. The effect of governing by throttling, and still more by "hit-and-miss," is to affect the composition even more strongly.

From the equation given above the ratio by volume of air to paraffin vapour for complete combustion is $\frac{116{\cdot}5+31}{2}$ or 74 to 1 as against about 45 to 1 for average petrol. On the other hand, paraffin weighs a good deal more per unit volume than petrol and on the whole there is little difference in output for a given cylinder capacity between the two. From tests that have been carried out it is pretty clear that the number of miles run per gallon of fuel on the same car is not very different whether petrol or paraffin is used.

**147. Road and Air Resistance.**—The performance of a petrol or paraffin engine as measured by the number of gross-ton-miles run by the vehicle per gallon of fuel depends upon the resistances encountered by the car in its motion. The total resisting force can be divided up into three parts, one independent of the speed, the second proportional to

the speed, and the third proportional to the square of the speed. Symbolically—

$$R=a+bv+Cv^2$$

where $R=$ total resistance in pounds

$v=$ speed of car in miles per hour

and $a$, $b$ and $c$ are constants.

The constant $a$ is chiefly related to the road resistance, the term $b$ to the frictional resistances of the mechanism, and the term $c$ to the air resistance and probably also to the throttling of the gases through the ports when the engine is racing. Experiments to show with accuracy what the values of the constants $a$, $b$ and $c$ should be, are lacking. With bicycles a good deal of work has been done, and Professor Perry many years ago gave the following formula for the total resistance to motion—

$$R=\frac{W}{80}\left(1+\frac{v}{20}+\frac{v^2}{200}\right)$$

where $R$ is measured in lb.; $W$ is the total moving weight in lb. and $v$ is speed in m.p.h. The author, as the result of some experiments carried out at Cambridge,* found

$$R=\frac{W}{132}\left(1+\frac{v}{20}+\frac{v^2}{84}\right).$$

The method by which this last-named formula was obtained was by noting the time taken to come to rest from a definite velocity when travelling over a flat road.† The same method could be used even more simply on a motor car by simply cutting off the spark and watching the speed indicator. No such experiments appear, however, to have been made.

**148.** Mr. Edge, however, has carried out some interesting tests at Brooklands to see what effect the raising of a large **wind screen** would have on the speed of a 6-cylinder Napier car giving 38·4 h.p. on the R.A.C. rating. His results were—

* *Mechanical Engineer*, November 11, 1899; *L'Ectairage Electrique*, November 30, 1901.

† In this case $T=\frac{2W}{fg}\tan^{-1}\frac{fv}{bv+2a}$ when $f=\sqrt{4ac-b^2}$.

| Area of Wind Resistance Screen in Square Feet. | Speed in M.P.H. |
|---|---|
| 30 | 47·85 |
| 28 | 50·0 |
| 26 | 52·9 |
| 24 | 56·15 |
| 22 | 54·0 |
| 20 | 55·5 |
| 18 | 57·0 |
| 16 | 57·6 |
| 14 | 60·0 |
| 12 | 62·5 |
| 10 | 64·2 |
| 8 | 66·15 |
| 6 | 70·25 |
| 4 | 75·0 |
| 2 | 73·8 |
| 0 | 79·0 |

These figures are shown plotted in Fig. 113. If it be assumed that the engine was driven during each run so as to give its maximum power it is interesting to work out the following exercise.

**149.** *Exercise.*—Let the $cv^2$ term in the resistance equation be written as $f(A+s)v^2$ where $A$ is the area of the wind screen, $s$ is the effective area of the front view of the car and $f$ is a constant.

Then $$R = a + bv + f(A+s)v^2.$$

Now the horse-power of output will be proportional to $R \times v$, and if it be assumed that the engine is always trying to exert approximately the same horse-power then

$$v \times \{a + bv + f(A+s)v^2\}$$

will be a constant quantity.

Now the value of this when $A=0$ is

$$79\{a + 79b + (79)^2 fs\}$$

To avoid too much arithmetic at this stage write $v_0$ in place of 79 and it follows that at all speeds the following relation must hold

$$v\{a + bv + f(A+s)v^2\} = v_0\{a + bv_0 + fsv_0^2\}$$

or $$v^3 + \frac{b}{f(A+s)}v^2 + \frac{a}{f(A+s)}v = \frac{v_0(a + bv_0 + fsv_0^2)}{f(A+s)}$$

a cubic equation in $v$.

As this equation only contains two variables $v$ and $A$, it shows directly how $v$ will vary for different values of $A$. If the constants $a$, $b$, $f$ and $s$ were known it would be possible to calculate the value of $v$ for, say, $A=30$ as in

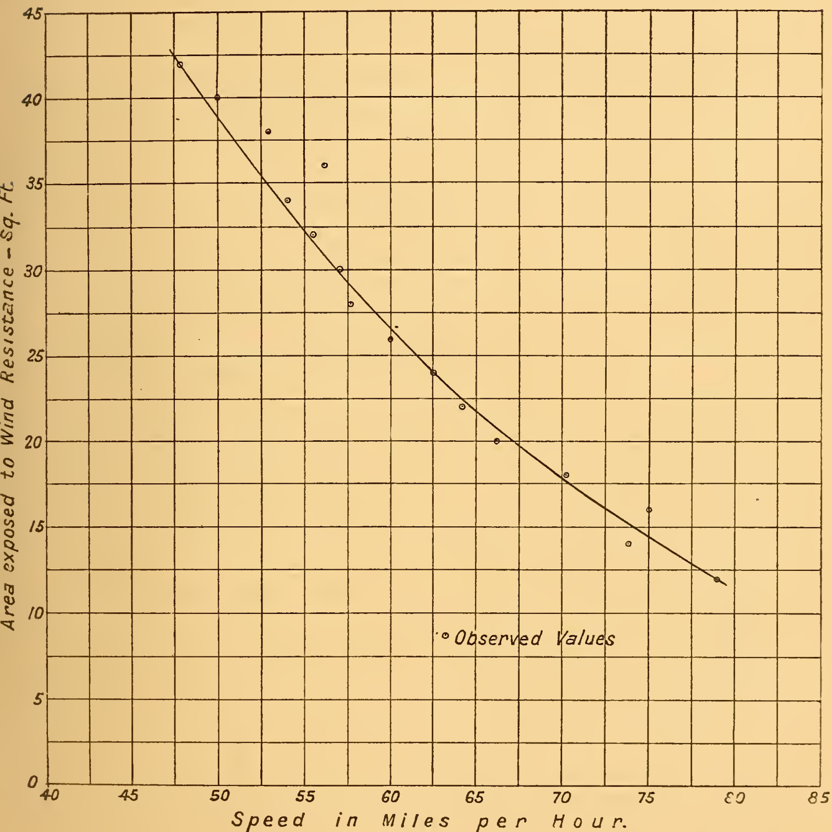



FIG. 113.—Diagram showing the effect of wind resistance on speed, as found by Mr. Edge on a Napier Car carrying a special wind screen.

the first line of the preceding table. If the constants were correct $v$ would come out at 47·85 m.p.h. as found by Mr. Edge. An interesting exercise that now arises for students to undertake is this. Reverse the above process. Given the curve (as in Fig. 113), find the values of the constants $a$, $b$, $f$ and $s$. As regards $s$ it may be pretty safely

assumed as about 12 square feet to start with. The author has not himself solved this exercise, nor, so far as he knows, has any one else. It is therefore possible for an enterprising student to do this and really find out for himself what are the constants in the equation

$$R = a + bv + cv^2$$

as applied to this car.

A good many suggestions have been made as to certain of the values of these constants taken alone. Thus the

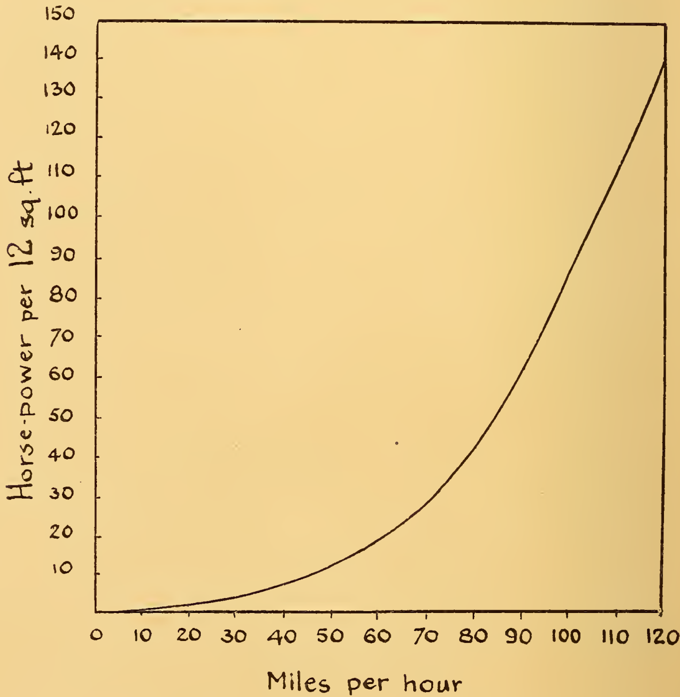



FIG. 114.—Showing relation between speed and H.P. necessary to overcome wind resistance per 12 sq. ft. of area.

road resistance is known to be not usually more than 50 lb. per ton, although cases of 100 lb. per ton are known. Also there are estimates of air resistance, and in Fig. 114 is

reproduced a curve based upon data given by Mr. Strickland in his valuable book on motor cars. He states that it is founded on information supplied by Col. Crompton.

**150.** *Exercise.*—If $v\{a+bv+f(A+s)v^2\} = v_0(a+bv_0+fsv_0^2)$ find the value of $\left(-\frac{dv}{dA}\right)$ i.e. the rate at which the speed is reduced as the area of the wind screen is increased.

Differentiate the above equation

$$\frac{dv}{dA}\{a+bv+f(A+s)v^2\}+v\{b\frac{dv}{dA}+2f(A+s)v\frac{dv}{dA}$$

$$+fv^2\}=0$$

or

$$\frac{dv}{dA}\{a+bv+f(A+s)v^2+bv+2f(A+s)v^2\}+fv^3=0$$

giving $\left(-\frac{dv}{dA}\right)=\frac{fv_3}{a+2bv+3f(A+s)v^2}$

When the student has worked out the values of $a$, $b$ and $f$ he can test the accuracy of this method of finding the slope of the curve in Fig. 113. Or he may, if he likes, proceed the other way round.

**151. Efficiency of Motor Vehicles.**—A standard of comparison early adopted was the measurement of miles run per gallon of fuel. This figure used to come out at about 40 for small touring cars and about 5 or 6 for the heavy vehicles used in commercial work. The great range of values of this standard made it difficult to compare cars with each other, and it is not difficult to see why this is so. The number of miles run per gallon is merely a measure of the distance through which the car is moved for a given number of thermal units supplied. To make the comparison real it is necessary to introduce some factor which takes account of the resistance overcome during the motion of the car. This resistance is proportional to a term of the form

$$(a+bv+cv^2)$$

where $v$ is the speed of the vehicle.

For slow-moving cars this expression is practically equal to $a$ and is a constant proportional to the total moving

weight. In other words the resistance overcome is proportional to the total moving weight. To express, therefore, the work done in moving the car over a given distance it is sufficient to multiply the distance travelled by the weight. In this way we arrive at the "gross-ton-miles-per-gallon" as a satisfactory standard of comparison. In the instances above given the gross-ton-miles-per-gallon would be about 30 for the small touring car and about 35 for the heavy one. That the touring car gives a lower result is due in part to the fact that in the case of rapid speeds (say over 20 miles per hour) air resistance as well as road resistance has to be allowed for, and that economy in fuel is more sought after in a commercial vehicle than in a pleasure one.

Very interesting series of trials of touring and commercial vehicles have been organized by the Royal Automobile Club, and from the figures given in their bulky reports the following tables have been made out—

1907 TRIALS OF COMMERCIAL VEHICLES.

| Net Load carried by Vehicles. | Average provided H.P. (R.A.C.) per ton of Gross Moving Load. | Average G.T.M. per Gallon of Petrol |
|---|---|---|
| ½ ton (5 cars) . . . | 7·82 | 27·4 |
| 1 ton (4 cars) . . . | 6·8 | 33·26 |
| 1½ tons (5 cars) . . . | 7·24 | 30·9 |
| 2 tons (7 cars) . . . | 4·96 | 34·93 |
| 3 tons (15 cars) . . . | 5·01 | 40·6 |
| 5 tons (1 car) . . . | 4·87 | 33·9 |

R.A.C. INTERNATIONAL TOURING CAR TRIALS, 1908.

| H.P. (R.A.C. rating) | Miles run per gallon (average of best cars). | Average weight (loaded) of best cars. Tons. | Gross-Ton-Miles per gallon of petrol. Average for best cars. |
|---|---|---|---|
| Up to 20 h.p. . | 24·2 | 1·30 | 30·7 |
| 20 to 40 h.p. . | 17·2 | 1·72 | 29·6 |
| 40 to 60 h.p. . | 15·9 | 2·14 | 33·9 |

*Note.*—Total distance run in the 1908 trials was 1,977 miles on the roads in Scotland and England. Average speed of all cars probably between 15 and 20 miles per hour.

These results, taken into consideration with others the author has had to deal with, suggest the figure of 35 as a good one to use as a convenient standard of performances of commercial vehicles when running on petrol—on paraffin, 30 is a fair figure to use. For touring cars it is less easy to quote a standard as the speeds have to be taken into account. One commercial car in the above trials (a Maudslay car fitted with the White and Poppe Carburettor, described in the previous chapter) achieved the phenomenal figure of 62, and it is of interest to see what such a good result means. It happens that the "G.T.M. per gallon" is really **a measure of thermal efficiency.** Thus if the road resistance be put at 50 lb. per ton and the number of foot-pounds stored up in each pound of petrol be 15,000,000, then the above performance shows a thermal efficiency of the whole car of

$$\frac{50 \times 62 \times 5{,}280}{7{\cdot}2 \times 15{,}000{,}000} = \text{about } 15 \text{ per cent.}$$

This means that of every 100 units of energy in the fuel fed into the engine no less than 15 units were transmitted to and used by the road wheels. Putting the loss between the crankshaft and the road wheels at 30 per cent. this corresponds to a ratio of $\dfrac{\text{B.H.P.}}{\text{thermal energy put in}}$ of 21 per cent., which is an exceedingly good figure seeing that it represents a general average of running at low and high speeds and all conditions of power output. With touring cars the figure of comparison is, as has been stated, in the neighbourhood of 26 (or 20 when running on paraffin) for a car of which the top speed is 20 miles per hour, going down to 20 as the maximum speed rises continuously to 40 miles per hour. Owing to the intervention of the air resistance at high speeds, the G.T.M. per gallon is not truly proportional to the thermal efficiency of the car. The thermal efficiency is really the G.T.M. per gallon multiplied by $\dfrac{a+bv+cv^2}{a}$ or $\left(1+\dfrac{b}{a}v+\dfrac{c}{a}v^2\right)$. The ratio of actual horse-power used to total weight moved should, of course, be

proportional to the velocity and could be used in calculating it, but in the first of the tables given on p. 318 the stated horse-power is only that which the engine could on certain suppositions exert as a maximum. It will be noted that the ratio falls as the load rises, so giving a general resemblance to the way in which the speed also falls as the vehicle gets heavier.

## PROBLEMS.

1. Describe, with sketches, an indicator which has been used to indicate petrol engines at 2,000 revolutions per minute. What were the results of Professor Callendar's investigations ? (B. of E., 1907.)

2. If a car is travelling along a level road find the time taken for the speed to fall when the ignition is switched off, from $v_0$ to $v_1$ and the space covered during this time.

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